THERMOACOUSTIC HEAT PUMPS

April 17, 1951

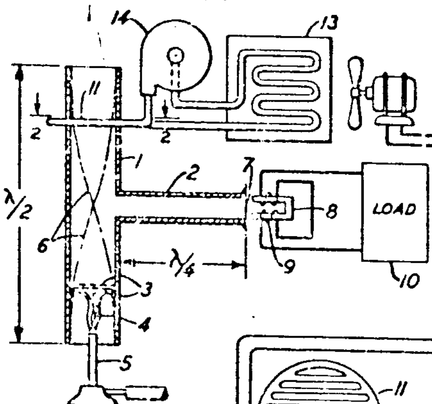
R. V. L. HARTLEY

2,549,464

ELECTRIC POWER SOURCE

Filed Oct. 29, 1947

FIG. 1



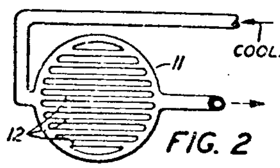


FIG. 2

FIG. 4

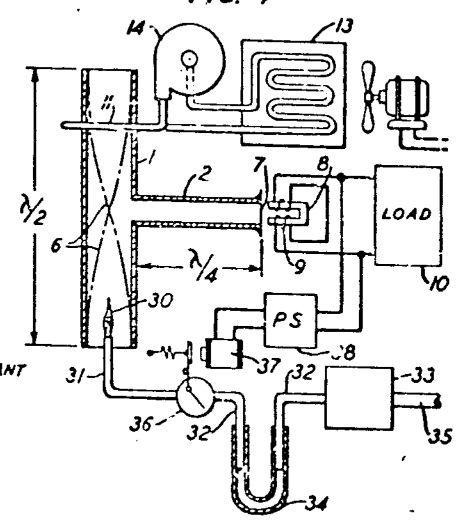


FIG. 3

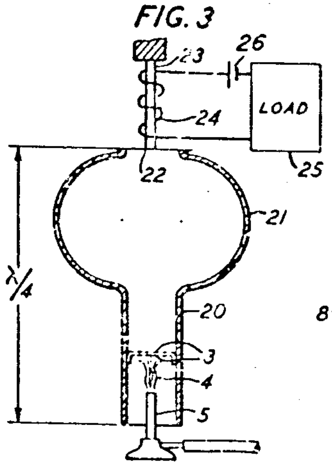
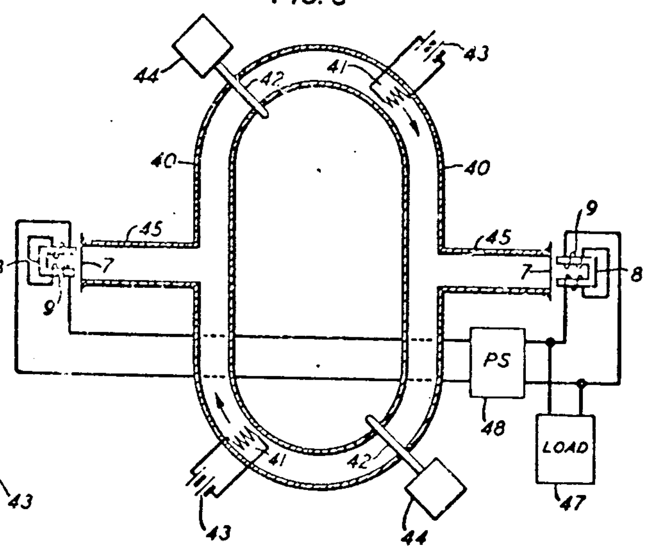


FIG. 5



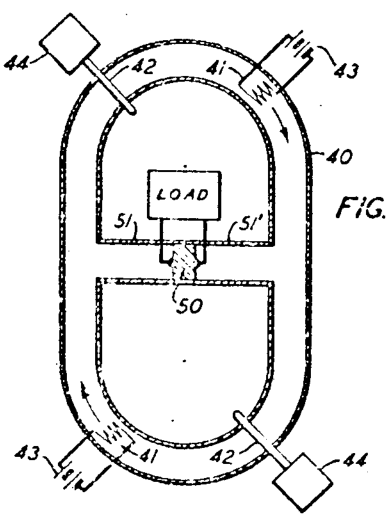


FIG. 6

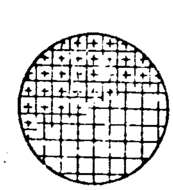


FIG. 7

INVENTOR
R. V. L. HARTLEY
BY Harry C. Hart
ATTORNEY

Patented Apr. 17, 1951

2,549,464

UNITED STATES PATENT OFFICE

2,549,464

ELECTRIC POWER SOURCE

Ralph V. L. Hartley, Summit, N. J., assignor to Bell Telephone Laboratories, Incorporated, New York, N. Y., a corporation of New York

Application October 29, 1947, Serial No. 782,703

1 Claim. (Cl. 290—1)

1

This invention relates to electric power sources.

A general object of the invention is to derive electrical energy from heat energy in a novel manner. A subsidiary object is to derive electrical energy from the energy of vibration of a fluid column. A related object is to improve the efficiency of thermo-acoustic vibrating systems.

It is known that compression waves of considerable amplitude may be set up in a column of air, gas, vapor or other elastic fluid confined in a pipe or tube, by the application of heat at a suitable point. In a simple form of the apparatus a cylindrical tube or pipe, open at both ends, is provided internally with a heating element, such as a screen of wire gauze, disposed approximately one-quarter of the distance from the lower end of the tube to the upper end. Upon application of heat, as by a torch, to the screen, the fluid column, which in this case is a mixture of air and gases, breaks into violent oscillations, emitting a loud roaring sound. In a modified arrangement, the upper end of the tube is terminated in a resonant bulb or chamber. In still another modification, the heat may be applied directly, as by a flame, to the enclosed fluid column. If preferred, the heat of the flame may be applied externally to the walls of the tube. The oscillatory pressure pattern remains fixed with respect to the tube despite the steady convective flow of the fluid through the tube and away from the heat source. Thus the wave pattern advances with respect to the medium with a velocity equal to that of the medium with respect to the tube. These devices have in the past served as educational exhibits for illustrating acoustical phenomena and relations, such as the dependence of the frequency and wavelength of the sound emitted on the length of the vibrating fluid column.

It is a feature of the invention that such apparatus is put to use as a generator of electrical energy. To this end a suitable mechanical-electrical transducer, such as a telephone receiver, a magnetrostrictive or a piezoelectric element is coupled, preferably by way of a suitable impedance matching transformer, to a suitable part of the vibrating fluid column. Compressional vibrations of the column are thus translated into electric currents which may then be supplied to a desired load. In addition, a part of the desired electrical energy may be fed back to control the timing of the application of heat, thus assisting in the maintenance of the oscillations.

In a modification, the tube is reentrant upon

2

itself, thus providing a closed path for a circuital fluid flow, and so conserving heat energy and increasing efficiency.

The invention will be fully apprehended from the following detailed description of preferred embodiments thereof, taken in conjunction with the appended drawings, in which:

Fig. 1 is a schematic diagram of apparatus in accordance with the invention in one of its simplest forms;

Fig. 2 is a cross-section of the apparatus of Fig. 1 taken at the section 2—2;

Fig. 3 is a schematic diagram of a modification of Fig. 1;

Fig. 4 is a schematic diagram of another modification of the apparatus of Fig. 1;

Figs. 5 and 6 are two alternative forms of a further modification of the apparatus of Fig. 1; and

Fig. 7 is a diagrammatic representation of a screen structure alternative to that of Fig. 1.

Referring now to Fig. 1, there is provided a tube or pipe 1, open at both ends and communicating with a second tube 2 at a point substantially midway between the two open ends of the first tube. Each tube may be of any suitable pressure resisting material, such as sheet metal or glass. The tube is preferably from 10 to 30 times its diameter. Thus, for example, it may be about 5 feet in length and 4 inches in diameter.

At a point intermediate the lower end of the tube 1 and its mid-point, for example, about one-fifth of the tube length from the lower end, there is provided a heating element which may comprise a screen of wire gauze, or a plurality of such screens 3 placed close to each other. This screen or screens may be shaped to fit the inside of the tube in any suitable manner and fixed in place in a plane approximately normal to the axis of the tube 1 either permanently or removably, as desired.

As is well known, upon the application of heat to the wire screen 3 by the flame 4 of a gas jet 5 or the like, the air or gas within the tube breaks into violent oscillations, standing compression waves are established and the tube emits a loud roaring sound. The wavelength of the fundamental component of the oscillations is approximately equal to twice the length of the tube. The mechanism by which this phenomenon takes place and whereby the steady heat energy applied to the screen 3 is converted into acoustical oscillatory energy has been many times described in the literature. In particular it is described in

"The Theory of Sound" by Lord Rayleigh, volume 2, pages 226–235.

While the oscillations are in progress, the oscillating velocity of the particles of air or gas is greatest at the two ends of the tube while the oscillating pressures are greatest at the mid-point of the tube. This condition is indicated by the broken lines 6 which show, approximately, the instantaneous velocities of the air particles at various points along the length of the tube at two successive half cycles of the oscillations, one half cycle apart.

In accordance with the invention, the pressure energy, which is greatest in the central region of the tube 1, is in part withdrawn to actuate a mechanical-electrical transducer. Thus a second tube 2 communicates with the vibrating air or gas column at the central point of the first tube 1. The second tube 2 is terminated in a vibratile diaphragm 7 which may be of conventional construction and which, when actuated by pressures within the tube, vibrates in the field of a permanent magnet 8, causing a current to flow in magnet winding 9 to which a load 10 may be connected.

Mechanical-electrical transducer elements of conventional form, including the telephone receiver schematically illustrated in Fig. 1, are characterised by impedances which, in the main, are high in comparison with the characteristic impedance of the vibrating air or gas column. Therefore, to withdraw substantial amounts of the energy of the vibrating column, some impedance matching device is desirable. A quarter wave impedance transformer is adequate for the purpose and accordingly the branch pipe 2 should be approximately one quarter wavelength from end to end, or, if preferred, an odd multiple of a quarter wavelength. The theory and operation of quarter wavelength transformers, both electrical and acoustical are well known per se and are described, for example, in "Electromechanical Transducers and Wave Filters" by W. P. Mason (Van Nostrand 1942). The quarter wavelength transformer or other impedance matching device is considered advantageous but is by no means essential, particularly because too perfect an impedance match between the vibrating air or gas column and the telephone receiver would result in too great a transfer of energy. If energy is withdrawn from the vibrating air column too rapidly the oscillations cannot be sustained.

It is to be understood that the pick-up device as shown in Fig. 1 may be replaced by any other suitable mechanical-electrical transducer, for example a piezoelectric element as in Fig. 6 or a magnetostrictive element as in Fig. 3.

The strength and the stability of the oscillations of the air or gas column are increased by the withdrawal of heat from a point intermediate the upper open end and the mid-point of this tube. To this end a cooling means 11 may be provided, for example, a screen of tubular members 12 as shown in Fig. 2, through the hollow meshes of which there flows a liquid such as water which may be circulated through a cooling radiator 13 external to the tube as by a pump 14.

Fig. 3 shows a modification of the invention in which the upper end of the tube 20 is closed by a bulb or resonator 21. Heat may be applied to a suitable point of the tube 20 by way of a heating element, such as wire gauze screens 3 as described in connection with Fig. 1. With this arrangement, high alternating pressures exist internally of the resonator 21 and the resonator wall therefore offers a suitable location for the

electromechanical transducer. Thus a vibratile diaphragm 22 may be mounted in an aperture in the resonator wall and a magnetostrictive member 23 provided with a winding 24 which may be connected to a load 25, may be mounted externally to the resonator so that when the diaphragm is actuated by pressures within the radiator, the member 23 is vibrated within the winding 24 and so modifies the current of a battery 26 flowing through the winding 24 to establish a varying current in a load 25.

Fig. 4 shows a further modification of the apparatus in which the wire gauze screens 3 and burners 4 of Figs. 1 and 3 are replaced by the flame 30 of a gas jet 31. This arrangement has the advantage that the heat may be applied intermittently and in synchronism with the oscillations of the air column in the tube 1, by alternately increasing and reducing the strength of the flame 30. Various means may be resorted to for this purpose and two are indicated in the figure. Thus the length of the gas pipe 32, measured from a reservoir 33 to the jet 31 may be altered as by adjustment of a U-tube 34 until this gas pipe is itself a resonant gas column having the same resonant frequency as the main tube. When this relation has been attained, and with suitable adjustment of the gas pressure in the gas main 35, then the alternate rise and fall of the pressure within the main tube 1 reacts on the jet 31 to cause alternate increase and reduction of the gas flow and therefore of the height and strength of the flame 30. Again, a valve 36 may be inserted in the gas pipe 32 which valve is alternately opened and partially closed in synchronism with the oscillations in the main tube 1. For example, the valve 36 may be operated by a relay 37 which is energized as shown. Electric current is supplied from the magnet winding 9, by way of an adjustable phase controlling device 38 to the relay 37. With this arrangement the strength of the oscillations in the main tube 1 and therefore the oscillatory energy which is available for withdrawal to the load circuit 10 is increased as compared with its value when the application of heat is steady.

With the open ended tubes of Figs. 1 and 4, some heat energy inevitably escapes to the atmosphere without contributing to the energy supplied to the load 10. To reduce these losses, the tube may be turned back on itself and made reentrant to provide a return path as indicated in Figs. 5 and 6. Thus in Fig. 5 there are two heating elements 41 and two cooling elements 42, so arranged around the endless tube 40 that the gas or air in its travel meets first an element of one type and then an element of the other. These elements are indicated schematically and each heater may be a wire screen or screens to which heat may be applied by external means, as by a battery 43. The cooling elements 42 may be of any desired type, for example, screens of tubular material as indicated in Figs. 1 and 2. They may be cooled by units 44 which may comprise pump, radiator and fan as in Fig. 1. The full length of the closed tube 40, measured along its axis around the circuit and back to the same point, is a full wavelength of the enclosed air or gas column or an integral number of full wavelengths. In operation, there is a small steady flow in one direction superposed on the rapid air or gas particle movement of oscillation. The direction of this movement depends on the arrangement and supply of the heating and cooling elements. Thus, if a large amount of heat is

applied to the heater 41 at the lower portion of the left-hand branch of Fig. 5 and a smaller amount at the upper end of the right-hand branch, there will be a net convective flow of the air or gas in the tube in a clockwise direction and oscillations will be sustained. If the heating elements are placed at the lower part of the right-hand branch and at the upper part of the left-hand branch with the cooling elements between them, and the heat supplied to the former element exceeds that supplied to the latter, the convective flow will be in a counterclockwise direction and operation will be otherwise the same.

Energy of vibration may be withdrawn from suitable high pressure points of the system, for example from the mid-points of both tube branches. To this end a branch pipe 45 which may be one-quarter wavelength long communicates with one vertical leg of the closed tube 40 substantially at its mid-point, and a similar branch pipe 45' communicates with the other vertical leg at its mid-point. Each branch pipe 45, 45' may be terminated in a mechanical-electrical transducer which may be of the electromagnetic type described above and including a diaphragm 7, a magnet 8 and windings 9, or of any other suitable type. The outputs of these two transducers may be together fed to a desired load 47 and they may be connected in parallel, although it is preferred to interpose an adjustable phase adjusting device 48 between them to compensate for any phase displacement which may exist between the oscillating pressures of the two branch pipes.

Fig. 6 shows a variant of Fig. 5 in which a single mechanical-electrical transducer, here shown by way of example as a piezoelectric element 50, is coupled to both branch pipes 51, 51' of the reentrant tube 40. In other respects the apparatus of Fig. 6 may be the same as that of Fig. 5, and its operation is similar. As in the case of Fig. 5, the heat supplied to the upward flowing branch should exceed that supplied to the downward flowing branch by an amount sufficient to sustain the steady convective flow in one particular direction and overcome the retarding effect of friction against the tube walls.

The closed reentrant tube arrangements of Fig. 5 and 6 lend themselves to use with the vapor of a high boiling point liquid such as mercury. This serves to facilitate the interchange of heat energy from the vibrating fluid column to the heater screens 41 and increases the efficiency of energy transfer to a high impedance mechanical-electrical transducer such as the piezoelectric element 50.

Wherever a wire gauze screen is employed, improved results may be obtained by the use of several such screens placed close together with their openings staggered so that as much as

possible of the fluid passing through the screen comes into immediate contact with one or other wire of the mesh, thus facilitating transfer of heat from the wire screen to the fluid. This construction is shown in Fig. 7.

What is claimed is:

A source of oscillatory energy comprising a tubular resonator, an elastic fluid within said resonator and capable of supporting standing compression waves of wavelength and frequency determined by the dimensions of said resonator, means for periodically applying heat at a selected point of said resonator to said fluid, whereby standing compression waves are developed in said fluid, means, including a mechanical-electrical transducer coupled to said fluid, for withdrawing a portion of the oscillatory energy of said compression waves, and means for feeding back a portion of the output of said transducer to control the timing of said periodic application of heat.

RALPH V. L. HARTLEY.

REFERENCES CITED

The following references are of record in the file of this patent:

UNITED STATES PATENTS

Number	Name	Date
782,146	Laudet	Feb. 7, 1905
1,213,611	Fessenden	Jan. 23, 1917
1,481,270	Purcell	Jan. 22, 1924
1,493,340	Hahnemann et al.	May 6, 1924
1,510,476	Hammond	Oct. 7, 1924
1,544,010	Jordan	June 30, 1925
1,795,647	Flanders	Mar. 10, 1931
1,954,516	Bourne	Apr. 10, 1934
1,969,037	Rieber	Aug. 7, 1934
2,017,744	Bourne	Oct. 15, 1935
2,075,263	Bourne	Mar. 30, 1937
2,094,621	Savage	Oct. 5, 1937
2,111,036	Wippel	Mar. 15, 1938
2,198,521	Whitelegg	Apr. 23, 1940
2,215,895	Wippel	Sept. 24, 1940
2,259,858	Reid	Oct. 21, 1941
2,278,668	Piety	Apr. 7, 1942
2,362,151	Ostenberg	Nov. 7, 1944
2,389,067	Lieberman	Nov. 13, 1945

FOREIGN PATENTS

Number	Country	Date
188,642	Great Britain	Nov. 29, 1923
424,955	Great Britain	May 31, 1933

OTHER REFERENCES

"The Theory of Sound," by Lord Rayleigh, volume 2, pages 226–235.

"Electromechanical Transducers and Wave Filters," by W. P. Mason (Van Nostrand, 1942).

May 27, 1958 W. A. MARRISON 2,836,033

HEAT-CONTROLLED ACOUSTIC WAVE SYSTEM

Filed July 15, 1953 4 Sheets-Sheet 1

FIG. 1

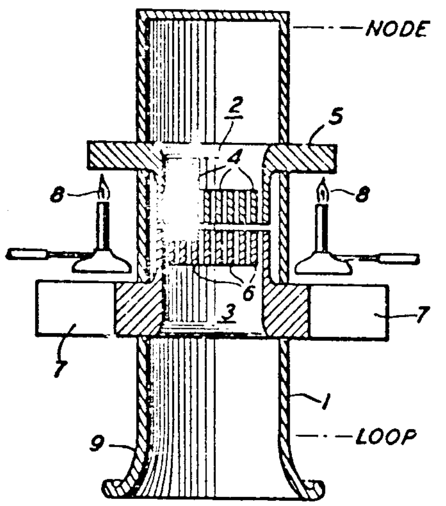


FIG. 2

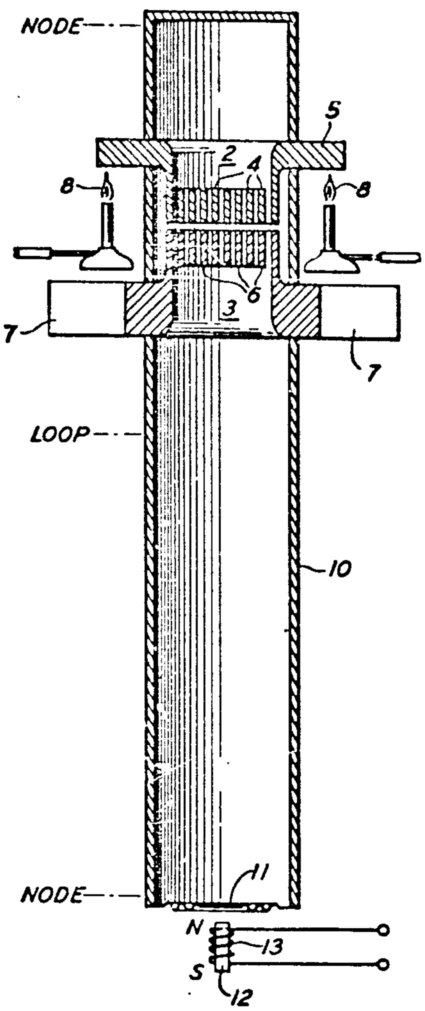
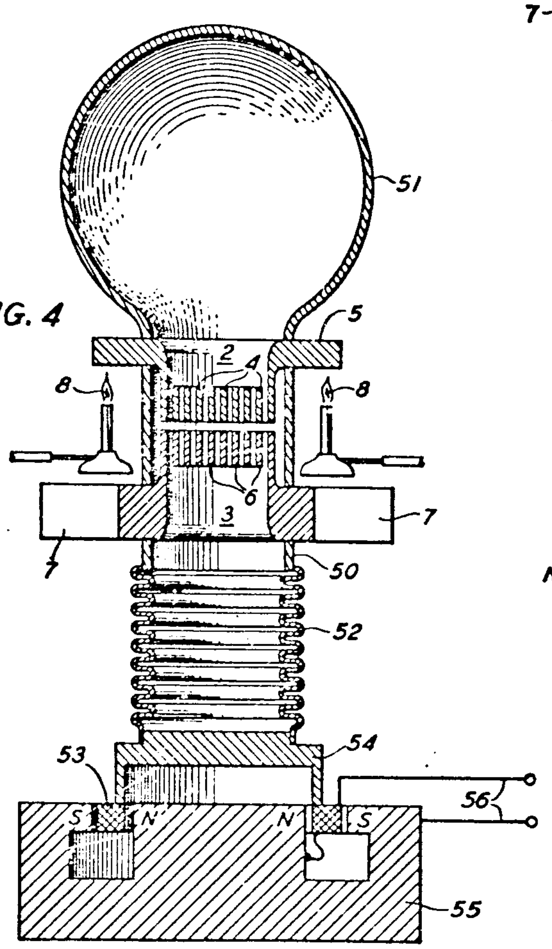


FIG. 4



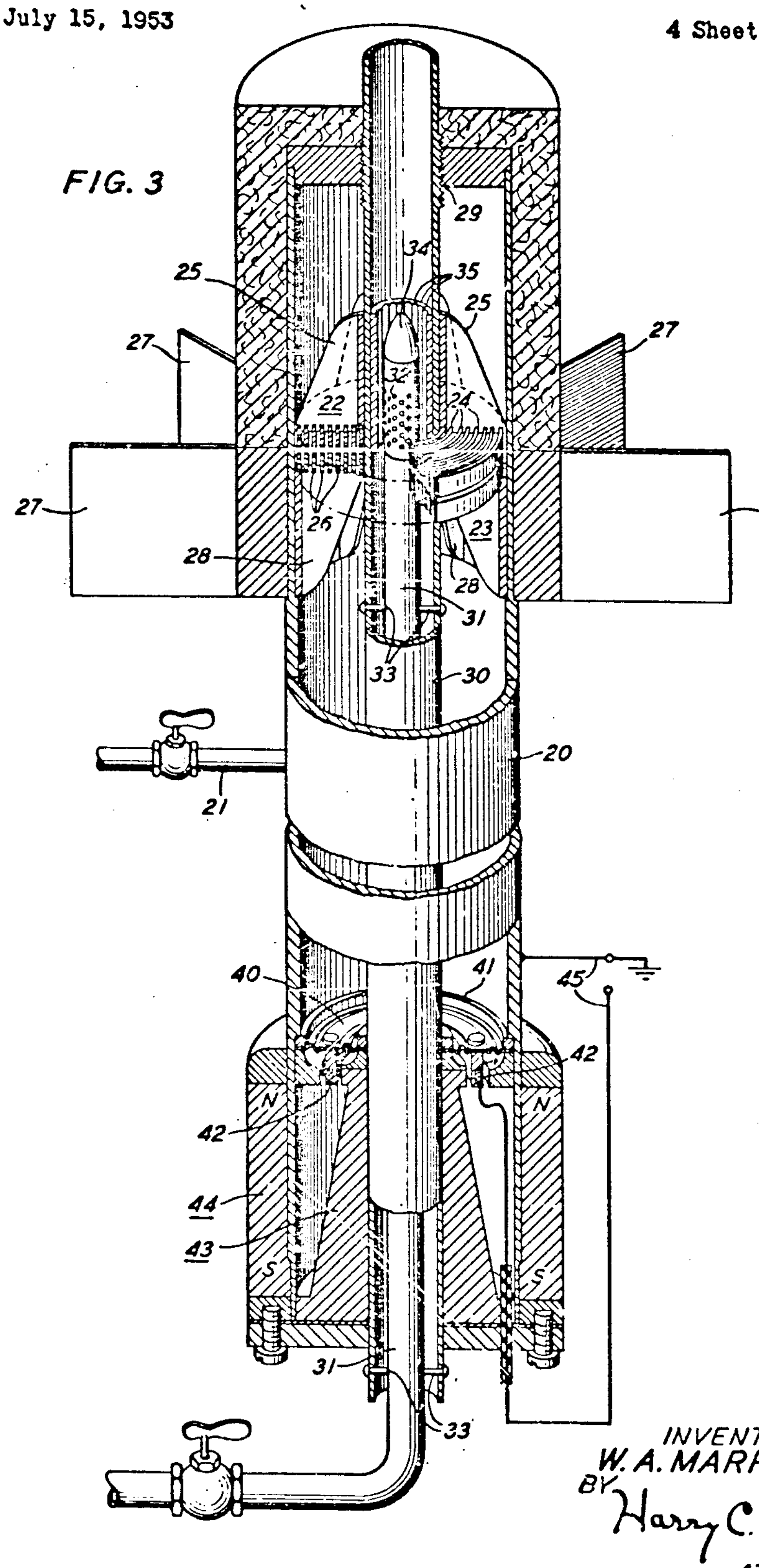
INVENTOR
W. A. MARRISON
BY Harry C. Hart
ATTORNEY

May 27, 1958 W. A. MARRISON 2,836,033

HEAT-CONTROLLED ACOUSTIC WAVE SYSTEM

Filed July 15, 1953 4 Sheets-Sheet 2

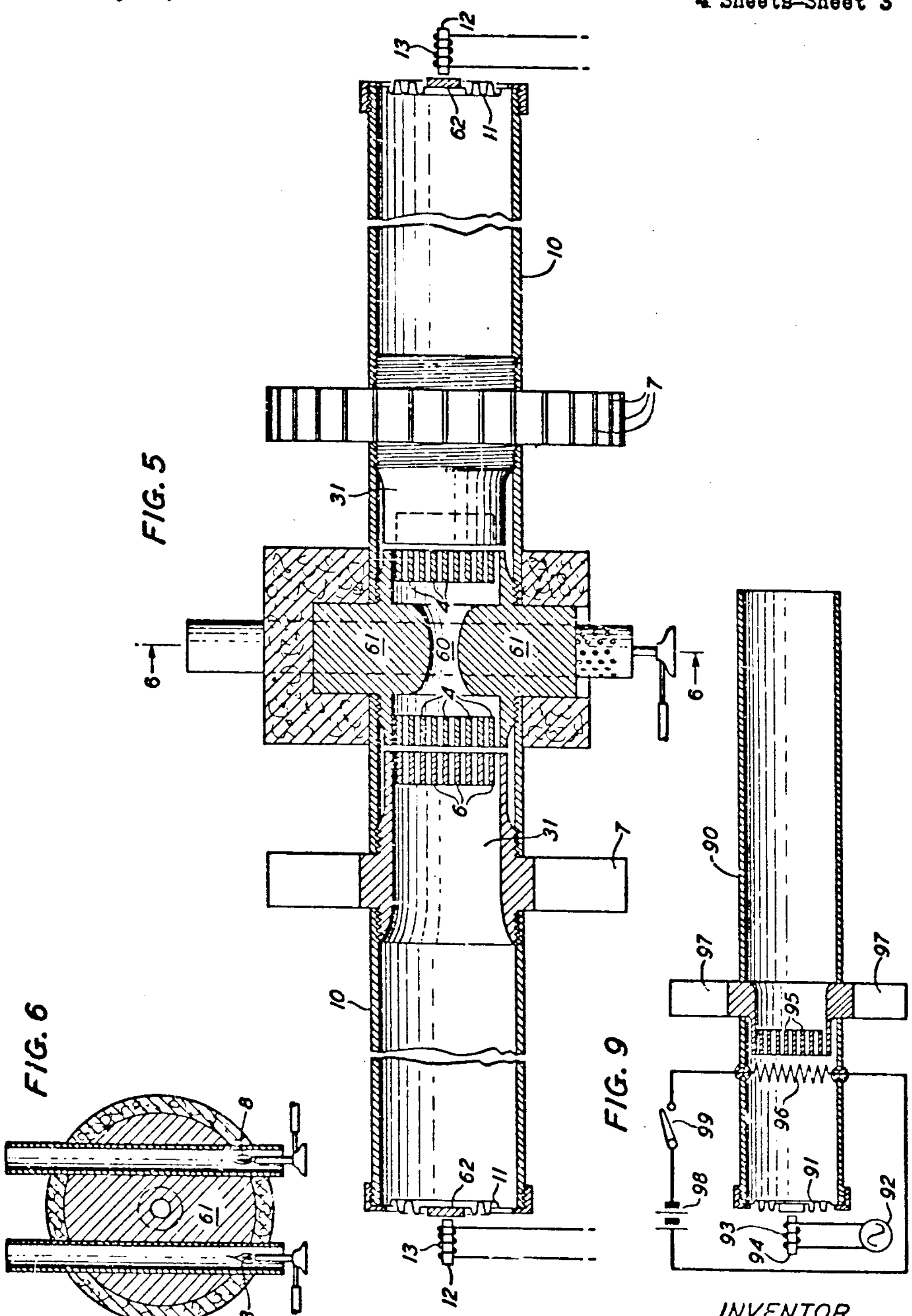
FIG. 3



INVENTOR
W. A. MARRISON
BY Harry C. Hart
ATTORNEY

HEAT-CONTROLLED ACOUSTIC WAVE SYSTEM

Filed July 15, 1953 4 Sheets-Sheet 3



INVENTOR
W. A. MARRISON
BY Harry C. Hart
ATTORNEY

FIG. 7

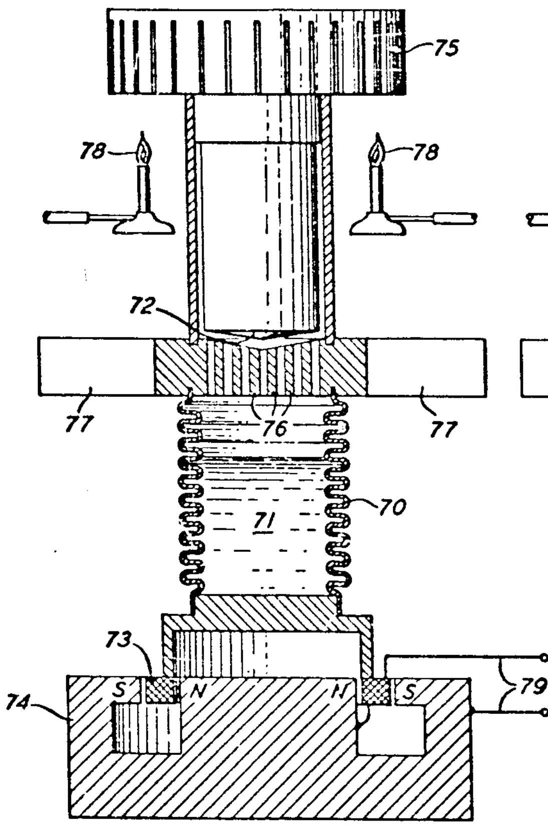
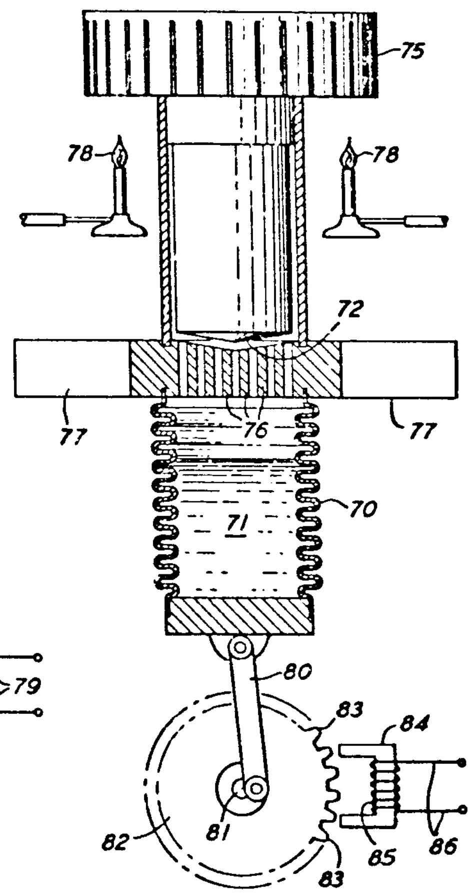


FIG. 8



INVENTOR
W. A. MARRISON
BY Harry C. Hart
ATTORNEY

United States Patent Office

2,836,033

Patented May 27, 1958

1

2,836,033

HEAT-CONTROLLED ACOUSTIC WAVE SYSTEM

Warren A. Marrison, Berkeley Heights, N. J., assignor to Bell Telephone Laboratories, Incorporated, New York, N. Y., a corporation of New York

Application July 15, 1953, Serial No. 368,185

23 Claims. (Cl. 60—24)

This invention relates to heat-controlled acoustic wave systems.

The principal object of the invention is to sustain acoustic oscillations in an elastic fluid as a working medium by the application of heat to it and the abstraction of heat from it in an effective manner. A related object is to amplify acoustic wave energy by such application and withdrawal of heat. A particular object is to convert the energy of burning fuel into energy of acoustic vibration without resort to rotating parts and with a minimum of movement of mechanical masses. Such acoustic vibratory energy is useful in many connections, for example in the supply of operating power to telephone apparatus at an unattended repeater station.

It is known that by the application of heat to one part of a confined column of an appropriate fluid, such as gas, a liquid or a vapor, and the extraction of heat from another part, the fluid column may be set into longitudinal vibration. It has already been proposed to convert the vibration energy into electrical energy by way of a transducer and to utilize this electrical energy in any desired fashion, for example to supply the bias voltages and currents required for the operation of an amplifier or other component of an unattended telephone repeater station. So far as is known, however, such apparatus is open to the objection that its efficiency is low.

The present invention is based in part upon the discovery that by the location of the element by way of which heat is supplied to the vibrating fluid column in optimum relation to the location of the element by way of which heat is withdrawn from it, a great improvement in efficiency may be obtained. In accordance with the invention in one of its principal forms, there is provided a container, for example a cylinder of length several times greater than its diameter, closed at one or both ends, and filled with a wave-supporting fluid medium, preferably a gas. When the column of fluid defined by a cylinder having one open end is set into longitudinal vibration, a vibration node exists at a closed end and a vibration loop exists at an open end. To the contrary, in the case of a cylinder in which both ends are closed, a node exists at each closed end and a loop exists at a distance removed from the node by a quarter wavelength. A heater member and a cooler member are located approximately midway between such a vibration node and such vibration loop. In particular, in the case of the column which is closed at both ends, these members are located approximately half way between one closed end of the column and its center. Each of these members has the form of a screen of strips which extend axially for a distance substantially equal to the amplitude of vibration of the fluid. They are equally spaced close together, indeed as close as possible without defeating the purpose of the invention by physical contact. The transfer of heat from one to the other by conduction and radiation is minimized by appropriate surface treatment. With this construction, each fluid molecule which passes through the heater-cooler pair is heated during substantially one half of its vibration cycle and cooled throughout substantially the other half. In accordance with a further feature of the invention, the heater member is located between the cooler member and the vibration node. With this arrangement the expansions and contractions of the gas which take place due to this alternating heating and cooling action are in such phase as to promote oscillation.

For operation in this fashion it is required that the phase of the transfer of heat from the heater member to the fluid moving into the heater member be somewhat lagging with respect to the physical displacement of this part of the medium to which this heat energy is imparted. With proper construction, this phase lag is obtained by virtue of the length of time required for the transfer of heat from the heater member to the working medium. In other words, as any fluid element passes into the heater member, the transfer of heat from the heater member to it commences immediately, but is not completed until after the fluid has completed its excursion and is on the way back to the cooler member.

Construction of the apparatus in accordance with the present invention optimizes this phase lag by providing for proper correlation of the dimensions of the heater member with the constants of the working medium and the oscillation period. Specifically, when the working medium is a gas, and provided the foregoing restrictions are also met, a satisfactory phase relation can be established when these constants are related in accordance with the following formula:

$$T=\frac{cdl^2}{2k}$$

where

T=the oscillation period
c=the mean specific heat of the gas
d=the density of the gas
l=the spacing between the strips
k=the thermal conductivity of the gas

all expressed in consistent units.

By the same token, the abstraction of heat from the working medium by the cooler member should lag the oscillatory displacement into the cooler member, provided the latter is located as described above with respect to the oscillation node and the heater member. Provided again that the working medium is a gas, the cooler member is preferably designed in accordance with the same formula.

In an elastic fluid which is undergoing longitudnal vibration, the velocity node is a point of maximum oscillatory pressure. More generally, therefore, the heater-cooler pair of the invention are disposed in the manner described above with respect to a point of maximum oscillatory pressure. In the case of an elastic fluid body which is driven into vibration by external means as distinguished from undergoing self-oscillatory vibrations, such a point of maximum pressure appears at the face of the driving member which may be a piston, the face of a piezoelectric crystal, an electromagnetically actuated diaphragm, or the like. In such a system the heater-cooler pair, when disposed in the fashion described above with respect to this point of maximum oscillatory pressure, may be regarded as furnishing negative resistance into which the driver works, and the apparatus which embodies them thus operates to amplify the acoustic energy applied by the driver to the medium.

Oscillations of large amplitude can in this way be produced. With an open-ended tube, a loud noise, which may serve as a fog horn or an alarm, is generated. More usually, however, an electrical output is desired, the noise being undesirable. For such purposes, a tube closed

at both ends and constructed of rigid material such as brass or steel is recommended. By the provision of a transducer of conventional design at an appropriate point of the apparatus, the energy of vibration of the fluid column may be converted into electrical energy for use as desired.

If desired, a pair of like units may be mounted in end-to-end relation and acoustic coupling may be provided to maintain their vibrations in dynamic balance.

The invention is in part also applicable to a heat engine of the 2-phase type, e. g., a liquid phase and a vapor phase. In such apparatus the vapor phase is caused to expand and contract due to alternate vaporization of a small fraction of the liquid and recondensation thereof. These expansions and contractions operate to move a portion of the liquid in bulk and this movement may be converted into reciprocating or rotary movements by known mechanisms. By the location of a cooling element in the optimum position relatively to the point at which heat is applied to the vapor, improved performance of such apparatus ensues.

The invention will be fully apprehended from the following detailed description of preferred embodiments thereof taken in connection with the appended drawings in which:

Fig. 1 is a diagrammatic cross-sectional view of a noise generator in accordance with the invention;

Fig. 2 is a diagrammatic cross-sectional view of a power generator in accordance with the invention, reduced to simplest form;

Fig. 3 is a diagrammatic cross-sectional view of preferred apparatus in accordance with the invention;

Fig. 4 is a diagrammatic cross-sectional view of a variant of the generator of Fig. 3;

Fig. 5 is a diagrammatic cross-sectional view of a pair of generators in accordance with the invention coupled together in end-to-end fashion for balanced operation;

Fig. 6 is a sectional view of the apparatus of Fig. 5 taken at the section 6—6;

Fig. 7 is a diagrammatic cross-sectional view of a modification of the invention;

Fig. 8 is a diagrammatic view of a variant of the apparatus of Fig. 7; and

Fig. 9 shows an acoustic amplifier embodying the principles of the invention.

Referring now to the drawings, Fig. 1 shows a cylindrical tube 1 of rigid material such as steel or brass pipe. Its upper end is closed and its lower end is open. Approximately midway between its upper end and its lower end there are provided a heater member 2 and a cooler member 3, the heater being located above the cooler, that is, between the cooler and the closed end of the tube. Each of these members may comprise a grid or screen of strips of sheet metal disposed parallel with each other and equally spaced apart, and aligned with those of the other member and axially of the tube in such a fashion as to provide the freest possible passage of a fluid entirely through both members. The strips 4 of the heater member 2 are conductively fixed as by welding to a flange 5 which extends outward through the walls of the cylinder. The strips 6 of the cooler member are similarly fixed in heat-conductive fashion to cooling fins 7 which likewise extend outwardly through the cylinder walls. Heat is applied to the flange 5 as by way of gas burner flames 8. The heat travels to the metal strips 4 of the heater member 2 by metallic conduction and is transferred to the gas within the tube by reason of gaseous conduction and its motion past the surfaces of these strips in the course of its vibrations. In reverse fashion, heat is withdrawn from the gas which passes between the metal strips 6 of the cooler member 3 and is withdrawn by metallic conduction to the cooling vanes 7 where it is conveyed to the exterior of the apparatus by conduction and convection to the atmosphere.

Application of heat to a device of the proportions

shown results in the establishment of longitudinal vibrations within the tube 1 which are of substantial amplitude. The wavelength of such vibrations is four times that of the tube 1 itself, a node appearing at the closed upper end and a loop at the open lower end. In other words the tube is one quarter wavelength long. It might equally well have a length of three, five, or any odd number of quarter wavelengths, with appropriate terminations at the ends of the tubes. To match the impedance of the device to that of the air and so transfer a maximum amount of power, the open end of the tube 1 may be provided with a bell or horn 9. This apparatus generates a loud noise which may serve as an alarm, a fog horn, or the like.

If preferred, the known principles of organ pipe design may be followed in the construction of a pipe which is highly resonant to a particular preassigned frequency and resonant to harmonics of this frequency to a desired extent. As so constructed the acoustic oscillator of Fig. 1 gives a musical tone of preassigned pitch and quality, and the apparatus may be employed as an organ pipe. In this event, it may be desirable to provide for the rapid starting and stopping of the oscillations, to which end an electrical heater element in the form of a grid of resistance wires may be preferred. Application of electric energy from a battery or other suitable source by manual control of a switch acts to set the pipe into oscillation and to terminate such oscillations as desired. Such a system is shown in Fig. 9, which is described more fully below.

Fig. 2 shows a tube 10 which may be of the same general construction as the tube of Fig. 1, but of approximately twice its length and closed at both ends. It is filled with a suitable fluid, preferably an inert gas such as argon, neon, or helium, or a mixture of such gases. In accordance with the known principles which govern standing acoustic waves in closed cylinders, such apparatus sustains longitudinal fluid vibrations in which a node exists at each end and a loop midway between the two ends. To maintain these vibrations, a heater member 2 and a cooler member 3 which may be of the same construction as those of Fig. 1 are located from one third to one half of the way from the node to the loop, the heater being again above the cooler. In this case, the length of the tube is one half the vibration wavelength. It might equally be any multiple of a half wavelength. Because the fully closed apparatus of Fig. 2 conserves energy more than the open-ended tube of Fig. 1, longitudinal vibrations of even greater amplitude may be established. To convert their energy into electrical energy, it is only necessary to provide a transducer of any desired variety and locate it at a point of the vibrating gas column which is appropriate from the standpoint of impedance matching. The vibration node is a high impedance point and for most purposes it is preferable to draw the output from such a point. Accordingly, a transducer of the simplest variety, namely, of a diaphragm 11 of magnetizable material, hermetically sealed to the walls of the tube 10 and providing one end closure for the tube, is associated with a magnetized ferromagnetic core 12 on which is wound a coil of wire 13. As the diaphragm 11 moves under the influence of the fluid vibrations, electrical energy is generated in the coil which may be applied to any desired load.

Fig. 3 shows the constructional details of apparatus which embodies the foregoing principles as well as certain refinements thereof. As before, a cylindrical tube 20 of rigid material such as brass or steel is provided. Within this tube is a second tube 30 of smaller diameter. The outer tube 20 is closed at each end by a ring whose inner diameter fits snugly around the wall of the inner tube, thus leaving the inner tube 30 open at both ends. Within the inner tube is a third tube 31 which serves merely to carry fuel such as illuminating gas or a combustible vapor mixture, or a mixture thereof with air,

to a burner 32. The tube 31 may be held in coaxial alignment with the tube 30 by pins 33. The burner 32 may conveniently comprise a plurality of small holes which pierce the wall of the tube 31 through which the fuel passes. Additional air for combustion may pass upward inside of the tube 30 and outside of the tube 31. With this construction a very hot flame surrounds the burner 32, and the body of the burner itself glows red hot. Loss of its heat in the upward direction may be minimized by a white ceramic cap 34, mounted on top of the burner.

To receive and absorb the heat of the flame to maximum extent, the inner wall of the tube 30 may be provided, especially in the neighborhood of the burner 32, with a number of indentations or corrugations 35 which act greatly to increase its heat-absorptive surface as compared with that of a smooth-walled tube of the same mean diameter. This corrugated surface is fixed, by way of the tube walls, to upper extensions 25 of the heater member 22 in heat conductive fashion as by welding, soldering, or otherwise. Each of these upper extensions is similarly conductively fixed to a number of concentric rings 24 of sheet metal which serve as the vanes by way of which the heat of the flame of the burner 32 is ultimately delivered to the vibrating fluid.

Similar rings 26 are located close to, and in alignment with, the heater rings 24 and immediately below them. These cooler rings are similarly fixed to flanges 28 and these flanges in turn are fixed to the walls of the outer tube 20 and to cooling vanes 27 which extend radially outward therefrom.

Precise adjustment of the distance separating the heater rings 24 from the cooler rings 26 may be secured by coupling the inner tube to the outer tube at some point, e. g., the upper ring closure, by way of a screw thread 29. Rotation of the inner tube 30 with respect to the outer one 20 then acts to increase or diminish the axial spacing of one set of rings 24 from the other 26 and thus to adjust their separation to the optimum value. Heat transfer by radiation from one set of rings to the other may be minimized by polishing the ring surfaces, especially those which face each other.

For optimum results, the full distance from the node end of the heater to the loop end of the cooler should be substantially equal to the full peak-to-peak excursion of the vibrating gas molecules in the vicinity of the heater-cooler pair. With a tube of about 30 centimeters total length, closed at each end, molecular peak-to-peak excursions as great as one centimeter are obtainable. When the heater-cooler pair are located as recommended, approximately midway between the node and the loop, the peak-to-peak excursion at this point of the standing wave is reduced to about 7 millimeters. Therefore, allowing for a spacing between the heater and the cooler of not more than one millimeter, the axial depths of the heater strips and of the cooler strips should both be about 3 millimeters.

The lower end of the annular vibrating column is terminated by an annular diaphragm 40 whose outer periphery is fixed to the outer tube 20 while its inner one is coupled snugly to the inner tube in rotatable fashion. A small passage 41 may be provided in the outer retaining ring, leading from the upper face of the diaphragm to the lower one. It serves to equalize pressures on the two sides and so to prevent static deflection. A coil 42 is fixed to the diaphragm 40 in position to move freely in the annular air gap of the yoke 43 of a magnet 44. With this construction, vibration of the annular gas column contained between the tube 20 and the tube 30 actuates the diaphragm 40 and an electrical voltage appears at the terminals 45 of the coil, and this may be supplied to any desired load.

A pipe fitting 21 serves to admit the vibrating medium into the annular space between the outer tube 20 and the inner tube 30, or to withdraw it. This medium is preferably an inert gas such as helium, argon, or neon, or a mixture of such gases. Variation of the proportions of such components of different molecular weights varies the frequency of vibration. Inertness is desirable to prevent corrosion of metal surfaces. For the same reason it is desirable to exclude all oxygen and water vapor from the annular chamber.

The working medium may, if desired, be operated at pressures substantially in excess of atmospheric pressure. Higher pressure results in higher vibratory energy per unit of volume, and therefore in a higher power output for apparatus of the same overall size.

The arrangement of heater and cooler elements described above in connection with Figs. 1, 2 and 3 may also be applied to an energy source in which the elasticity function is largely concentrated at one end while the inertial or mass function is largely concentrated at the other end. Fig. 4 shows such a construction in which a tube 50 is provided at its upper end with an enlargement or bulb 51 and at its lower end with an expansible bellows 52. Between the bulb 51 and the bellows 52 a heater-cooler pair may be mounted having identically the construction of these elements in Figs. 1 and 2, like parts being similarly numbered. A coil 53 may be mounted by way of a bracket 54 on the lower end of the bellows 52 in position to travel vertically in the air gap of a magnet 55.

When heat is applied as by way of a flame to the heater element 2, the gas within the entire structure is set into vibration and the resulting oscillatory pressures cause the movable coil 53 to be reciprocated in the air gap of the magnet 55. This movement results in the generation of an electric voltage which appears at the terminals 56. It may be utilized as desired.

Fig. 5 shows a pair of devices of the type shown in Fig. 2 juxtaposed in end-to-end relation for dynamic balance. In order that there shall be adequate coupling between the two vibrating gas columns, a channel 60 is provided which interconnects them. The ends of the two columns which are spaced most nearly together would be vibration nodes were it not for the provision of this channel. The effective nodal plane for both columns is thus in the approximate center of the interconnecting channel 60. For this reason the heater-cooler pair is in each case spaced relatively close to the central plane. Heat may conveniently be applied to a heater 61, which is common to both tubes 10, by way of burners 8 as indicated in the cross-sectional view of Fig. 6.

In the operation of this balanced arrangement, the movement of the gas at any instant is either inward toward the central plane of both cylinders 10 or outward toward the separated ends of the two cylinders. Thus, the vibration of each gas column finds a reaction in the vibration of the other gas column so that dynamic balance is secured and external vibration, shaking of the mount, noise, and the like, are minimized.

The energy of each vibrating column may be withdrawn by way of a transducer mounted near a vibration node. Two such transducers are shown, one at each of the separated ends. Each may comprise a diaphragm 11, having mounted on it a magnetizable slug 62, juxtaposed with a core 12 on which is wound a coil 13.

Their electrical outputs, derived in the fashion heretofore described in connection with Fig. 2, may be connected in parallel or in series, as desired, and supplied to any desired load.

Fig. 7 shows another form of resonant heat engine in which a closed chamber is itself constructed principally of an expansible bellows 70. The chamber is largely filled with a vaporizable liquid 71; e. g., an ether, a saturated hydrocarbon, or, aside from the problem of corrosion, water. The upper surface 72 of the chamber is heated as by the application of heat from a gas flame 78 to the outwardly extending flange 75 which is integral therewith. This upper surface 72 is preferably pitted or corrugated to increase the area of its surface

as compared with that of a flat surface of the same overall extent. A grid or screen of sheet metal strips 76 is provided between the upper surface of the liquid 71 and the upper bound 72 of this chamber, and these strips are conductively fixed in suitable fashion to cooling fins 77 which extend outwardly of the apparatus. In operation, a portion of the liquid is alternately vaporized by coming in contact with the hot upper bound 72 of the chamber and condensed by coming in contact with the cooling strips 76. This results in alternate expansion and contraction of the vapor which lies above the liquid surface and so alternately drives the liquid 71 downward, extending the bellows 70, and upward, compressing the bellows. A coil 73 may be fixed to the lower end of the bellows 70 and may be arranged to move in the air gap of a permanent magnet 74, thus to generate electrical energy which may be withdrawn from the coil terminals 79.

Fig. shows a modification of Fig. 7 which may be of the ne construction and, insofar as the heat engine portion ereof is concerned, operates in the same fashion. Instead, however, of operating to reciprocate a winding in the air gap of a magnet, extension and retraction of the bellows 70 operates to advance and withdraw a connecting rod 80 and so to rotate a shaft 81 which bears a toothed wheel 82. Movement of the teeth 83 past the poles of a magnet 84 operates to generate electrical energy in a coil 85. This electrical energy may be withdrawn from the terminals 86 for use as desired.

Every self-oscillating system may be regarded as embodying an amplification principle and a feedback principle by which energy is fed back from the output of the amplifier to its input. The apparatus with which the present invention deals is no exception, and so embodies an acoustic amplifier. Such an amplifier is schematically illustrated in Fig. 9 which shows a tube 90 having an open end and an end which is closed by a diaphragm 91 which in turn is caused to oscillate by application of the signal of an alternating source 92 to a coil 93 wound on a core 94. A heater-cooler pair are provided between the diaphragm 91 where an approximate vibration node exists, and the location of the nearest vibration loop. The cooler may have substantially the same construction as in the other figures, i. e., metal strips 95 connected to external fins 97. The same may be true of the heater but for illustrative purposes it is preferred to employ as a heater a screen or grid 96 of resistance wire which is heated by application thereto of the current of a battery 98 when and if a switch 99 is closed. Because the successive application of heat to the vibrating medium, in this case air, and extraction of heat from it as it vibrates through and past the heater and the cooler are in such phase as to tend to amplify the impressed acoustic oscillations, the apparatus acts as an acoustic amplifier for acoustic energy applied to its closed end by vibration of the diaphragm 91. The apparatus of Fig. 9 may find utility as an electrically driven organ pipe.

Other applications of the foregoing principles will suggest themselves to those skilled in the art.

What is claimed is:

1. An acoustic wave system which comprises a vessel having fixed side walls and at least one fixed end wall and thus defining a constant volume, an elastic fluid wave-supporting medium contained within said vessel, a pair of closely spaced heat exchangers disposed within said vessel, means for applying heat to one of said exchangers, means for withdrawing heat from the other of said exchangers, and means for abstracting oscillatory energy from acoustic oscillations of said medium.

2. An acoustic wave system which comprises a vessel having fixed side walls and at least one fixed end wall and thus defining a constant volume, an elastic fluid medium contained within said vessel, said medium being free to undergo oscillatory flow in one direction within said vessel with a preassigned oscillation wavelength, a pair of heat

exchangers disposed within said vessel, said exchangers being spaced apart in the direction of fluid flow by a small fraction of said oscillation wavelength, means for applying heat to one of said exchangers, means for withdrawing heat from the other of said exchangers, and means for abstracting oscillatory energy from acoustic oscillations of said medium.

3. An acoustic wave system which comprises a vessel having fixed side walls and at least one fixed end wall and thus defining a constant volume, an elastic fluid wave-supporting medium within said vessel, said medium being free to undergo oscillatory flow in one direction within said vessel with a determinable amplitude, a pair of closely spaced heat exchangers disposed within said vessel, the dimension of each of said exchangers in the direction of said fluid flow being approximately equal to said oscillation amplitude, means for applying heat to one of said exchangers, means for withdrawing heat from the other of said exchangers at a preassigned rate, whereby said oscillation amplitude is determined, and means for abstracting oscillatory energy from acoustic oscillations of said medium.

4. Apparatus as defined in claim 3 wherein at least one of said exchangers comprises a plurality of strips which are equally spaced apart in a direction normal to said fluid flow and wherein said strip spacing is related to the specific heat, the density and the thermal conductivity, respectively, of the fluid medium and to the oscillation period in accordance with the following formula:

$$T=\frac{cdl^2}{2k}$$

where

T=the oscillation period
c=the mean specific heat of the medium
d=the density of the medium
l=the spacing between the strips
k=the thermal conductivity of the medium

5. An acoustic wave system which comprises a vessel having fixed side walls and at least one fixed end wall and thus defining a constant volume, an elastic fluid medium contained within said vessel, said medium being free to undergo compressional vibrations and so to have standing waves established therein, said waves being characterized by a velocity node at one part of said vessel and by a velocity loop at another part of said vessel, a pair of closely spaced heat exchangers disposed within said vessel, means for applying heat to one of said exchangers, means for withdrawing heat from the other of said exchangers, and means for abstracting oscillatory energy from acoustic oscillations of said medium.

6. Apparatus as defined in claim 5 wherein the heat exchangers are located between a velocity node and a velocity loop for the acoustic oscillations.

7. Apparatus as defined in claim 5 wherein the heat exchangers are located substantially midway between a velocity node and a velocity loop for acoustic oscillations.

8. Apparatus as defined in claim 5 wherein the heat exchangers are located with the hot exchanger on that side of their midpoint which is nearest to the velocity node and with the cold exchanger on that side of their midpoint which is nearest the velocity loop.

9. A source of oscillatory energy which comprises a hollow resonator having at least one closed end, an elastic fluid medium within said resonator which is capable of supporting standing compression waves of wavelength and frequency determined by the dimensions of said resonator, said waves being characterized by a node portion of maximum oscillatory pressure and a loop portion of maximum oscillatory displacement and separated by substantially one quarter wavelength from said node portion, means for withdrawing heat from said fluid medium at a point approximately midway between said node portion and said loop portion, means for applying heat to

said fluid medium at a point between said node portion and the location of said heat-withdrawing means and closely adjacent to said heat withdrawing means, and means for abstracting a portion of the oscillatory energy of said compression waves.

10. Apparatus as defined in claim 9 wherein said hollow resonator is a rigid cylinder of length at least several times its diameter.

11. Apparatus as defined in claim 9 wherein said hollow resonator comprises a cylindrical body of rigid material having at one end thereof a cylindrical extension of expansible material and at the other end thereof a bulbous enlargement.

12. Apparatus as defined in claim 9 wherein said hollow resonator comprises a first cylinder of rigid material of length at least several times its diameter, a second cylinder of equal or greater length and less diameter disposed within and coaxially with said first cylinder, and a ring coupling similarly located ends of said cylinders thereby to define a cylindrical annular chamber for containing the working fluid medium.

13. Apparatus as defined in claim 9 wherein the elastic fluid medium is a gas.

14. Apparatus as defined in claim 9 wherein the elastic fluid medium is a mixture of gases of different atomic weights, the proportions of the several components of said mixture being selected to adjust the vibratory energy to a desired frequency.

15. Apparatus as defined in claim 9 wherein the elastic fluid medium is a vapor, and wherein a supply of the liquid of said vapor is provided within said resonator to ensure a sufficiency of said vapor.

16. Apparatus as defined in claim 9 wherein the combined axial depths of the heat-applying means and the heat-withdrawing means are substantially equal to the peak-to-peak excursion of molecules of the fluid medium at the location of said two means in the course of standing wave vibrations of said medium within said resonator.

17. An oscillatory heat engine which comprises a rigid hollow cylinder of length at least several times its diameter, a fixed closure at each end of said cylinder, a gas filling the interior of said cylinder, a permeable cooler member mounted in said cylinder substantially midway between one end of said cylinder and its midpoint, a permeable heater member mounted in said cylinder between said cooler member and said end of said cylinder and in close proximity to said cooler member, means for applying heat to said heater member, and means for withdrawing heat from said cooler member, whereby the gas column contained in said cylinder is set into longitudinal vibration, and means for utilizing the energy of vibration of said gas column.

18. An acoustic wave generator which comprises a tubular vessel having a fixed closed end and an open end and defining a substantially constant volume, an elastic fluid wave-supporting medium contained within said vessel and substantially filling said volume, a first heat exchanger disposed within said vessel and located substantially midway between its ends, means for abstracting heat from said first exchanger, a second heat exchanger disposed between the closed end of said vessel and said first heat exchanger and closely adjacent said first heat exchanger, and means for applying heat to said second exchanger, whereby acoustic vibrations are maintained within said vessel and are communicated to the external atmosphere by way of said open end.

19. An acoustic amplifier which comprises a tubular vessel having a fixed closed end and an open end and defining a substantially constant volume, an elastic fluid wave-supporting medium contained within said vessel and substantially filling said volume, a first heat exchanger disposed within said vessel and located between its ends, means for abstracting heat from said first exchanger, a second heat exchanger disposed between the closed end of said vessel and said first heat exchanger and closely adjacent said first heat exchanger, means for applying heat to said second exchanger, a source of a signal of a frequency to which the fluid medium contained within said vessel is resonant, and means for applying energy of said source as an oscillatory pressure to said medium by way of said closed end, whereby acoustic vibrations at said frequency are maintained and amplified within said vessel, and whereby said amplified vibrations are communicated to the external atmosphere by way of said open end.

20. A source of oscillatory energy which comprises two hollow tubular resonators, each having a completely closed end and a partially open end, an elastic fluid medium within each resonator which is capable of supporting standing compression waves of wavelength and frequency determined by the dimensions of said resonator, said waves being characterized by a node portion of maximum oscillatory pressure and a loop portion of maximum oscillatory displacement and separated by substantially one quarter wavelength from said node portion, means for applying heat to the fluid medium of each resonator at a point approximately midway between said node portion and said loop portion, means for withdrawing heat from the fluid medium of each resonator from a point between said loop portion and the location of said heat-applying means and closely adjacent to said heat-applying means, means for abstracting a portion of the oscillatory energy of the compression waves of each resonator, and a port interconnecting the partially open ends of said resonators, thereby to equalize the pressures within said resonators at their partially open ends and so to establish a pressure node at the midpoint of the system.

21. Apparatus as defined in claim 20 wherein said resonators are of like dimensions and wherein the fluid media contained therein are of like properties.

22. Apparatus as defined in claim 20 wherein said resonators are mounted on a common axis with their partially open ends adjoining one another.

23. In combination with apparatus as defined in claim 22, a source of heat common to the heat-applying means of said two resonators.

References Cited in the file of this patent

UNITED STATES PATENTS

2,215,895	Wippel	Sept. 24, 1940
2,368,101	Böttcher	Jan. 30, 1945
2,402,544	Foulds	June 25, 1946
2,471,832	McCollum	May 31, 1949
2,522,389	Mason	Sept. 12, 1950
2,535,680	Horsley et al.	Dec. 26, 1950
2,539,535	Espenschied	Jan. 30, 1951
2,549,464	Hartley	Apr. 17, 1951
2,560,987	Rinia et al.	July 17, 1951
2,596,051	Stigter	May 6, 1952
2,745,372	Chertoff	May 15, 1956

Advanced Cryogenic Engineering
1966, p. 171-79.

C-7

SURFACE HEAT PUMPING

W. E. Gifford and R. C. Longsworth

Syracuse University
Syracuse, New York

INTRODUCTION

Refrigeration has been of great interest to man for generations, mostly for food preservation involving relatively small temperature differences. Recently, however, there have evolved many other applications for a great variety of other temperatures. Despite all this recent historic interest, there are really very few methods by which heat may be removed and low temperatures achieved.

This paper summarizes a heat-pumping process which may be set up on any surface of a closed chamber where the pressure is varied by the delivery of gas from one point. This process can be used to build up much greater temperature differences and also exhibit higher efficiencies than some of the more widely known refrigeration methods. For example, the Pulse Tube cycle utilizes this effect in such a manner that it has been demonstrated to pump heat from one end of a 12-in.-long thin-walled tube at − 208°F to the other end at 72°F.

DESCRIPTION OF THE EFFECT

Surface heat pumping is caused by an unusual interaction between fluid displacement along a surface, energy change in the fluid, and heat exchange with the surface, as a result of a periodic change of pressure of the gas. The mechanism can best be visualized from consideration of the closed tubular model shown in Fig. 1. A small element of gas at X' with temperature T_1' is displaced to X'' as a result of pressurizing the tube by supplying gas from the left end. Also, as a result of pressurizing the element of gas there is an energy increase in the gas which produces a change in the temperature of the gas. It is possible that the element will be in thermal equilibrium with the wall but more likely the gas will be compressed along a polytropic path, or if the pressure change is fast enough so that essentially no heat is transferred from the element, then an isentropic path is followed, as shown by the dashed line in Fig. 1, reaching a temperature of T_1'. If the pressure in the tube is now held constant, heat will flow from the gas to the wall, eventually cooling to T_2'. When gas is now allowed to flow out of the tube and the pressure reduced to its initial value the element will return to the vicinity of its initial position but will have a temperature T_2'' which is lower than T_1'. The cycle is completed by maintaining the pressure constant while heat flows from the wall to the gas, returning the temperature to T_1'.

The net effect of this cycle has been the removal of heat from the wall at X' and the depositing of that heat in the wall at X''. This effect occurs throughout the tube and produces a heat-pumping effect from the open end to the closed end.

While this effect occurs in any volume, there are several factors which enhance the effect. First, the flow should be smooth and uniform, so that a regular and not a random pattern of heat pumping is established. Second, the heat exchange with the walls when the gas is at its extreme positions should be large relative to the heat that is exchanged while it is moving.

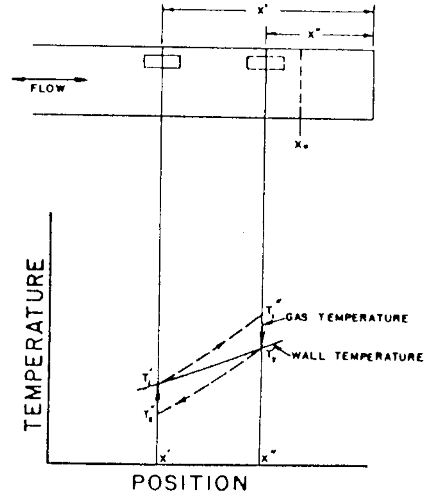


Fig. 1. Gas and wall temperature *vs.* position for surface heat-pumping cycle.

For the present it will be considered that there is no viscous drag from the surfaces, so that the gas flowing back and forth in the volume shown in Fig. 1 has a constant velocity across the cross section of the tube. If the pressure change occurs fast enough, heat transfer during the pressure change is negligible and the gas follows an isentropic path. The temperature is thus related to the pressure P according to the well-known thermodynamic relation,

$$T'' = T'\left(\frac{P''}{P'}\right)^{(\gamma-1)/\gamma} \tag{1}$$

in which the single prime denotes the initial condition and the double prime denotes the final condition.

The assumption of an isentropic process throughout the tube with uniform velocity across the cross section also permits the use of the relation between temperature and displacement.

$$T'' = T'\left(\frac{X'}{X''}\right)^{\gamma-1} \tag{2}$$

At any given point along the wall the temperature will assume some value between the temperature of the gas adjacent to it when the pressure is low and the temperature of the gas when the pressure is high. Heat will be exchanged with the wall and pumped along the wall until such a condition is reached that the gas temperature coincides with the wall temperature everywhere along the tube. If now the temperature at $X = X_0$ is held at T_0 (Fig. 1) and the temperatures in the rest of the tube are allowed to come to equilibrium with the gas, under the condition in which no heat is being pumped, then the temperature in the tube at X, $T(x)$ will be

$$\frac{T(x)}{T_0} = \left(\frac{X_0}{X}\right)^{\gamma-1} \tag{3}$$

This action will occur for virtually any pressure change which causes a displacement of the gas in the tube. Small pressure ratios of 1.5 to 2.0 will cause the mechanism which will give the heat-pumping action. As can be seen, the temperature changes built up in the walls depend on volume ratios in the tube. Therefore, the temperature differences it would be possible to build up in a tube would be very great relative to that defined by the pressure changes in (1).

As the temperature pattern gets near to that defined by (3) the temperature differences become smaller and smaller and the heat-pumping action is greatly reduced. One might

expect to approach this temperature pattern, but in an actual tube there would be some conduction and heat exchange losses which would prevent one from actually reaching it.

The effect of the viscous drag of the tube surfaces on the heat-pumping action can best be visualized by again considering the oscillating flow in the same tube. Due to the fact that the walls cause a viscous drag on the gas, the flow will be slower at the walls and faster in the center when the pressure oscillations are instigated. As a result, the gas near the walls will not be displaced as far by the pressure oscillations as the gas in the center of the tube.

To see how this variation in gas displacement will affect the heat-pumping phenomenon, assume that the tube has the ideal temperature pattern in it defined by (3) and shown in Fig. 2. In such a case the gas that is displaced the average distance will not have a temperature difference which will cause a heat transfer with the walls. As this gas is displaced its temperature will coincide with that of the wall. Thus, this gas would have no tendency to create a heat-pumping action.

The gas near the surfaces, however, will vary as much in temperature as the average gas but will not move so far. It will only be displaced from X_1' to X_1'', instead of the average displacement from X_2' to X_2'', as shown in Fig. 2. As a result it will be warmer than the tube at X_1'' and colder at X_1'. It will, therefore, heat the tube at X_1'' with cooling of the

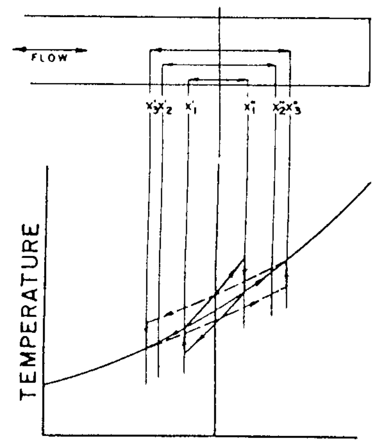


Fig. 2. Gas and wall temperature *vs.* position for elements at different radii with viscous effects.

gas and cool the tube at X_1' with heating of the gas. The net result is the transfer of heat from X_1' to X_1'' against a temperature gradient. This action is, of course, true for every section of the tube. There is thus a very positive heat-pumping action at the surface even when the effect described, assuming no viscosity, is no longer possible.

At the center of the tube, gas will be displaced further than the average, between X_3' and X_3'', and will build up temperature differences with respect to the walls, thereby tending to pump heat in the opposite direction if there were heat transfer between this gas and the walls. However, the walls are not available to the central gas, and thus its temperature will just rise and fall with the displacements in an essentially reversible manner.

It is quite interesting that the viscous drag of the surfaces distorts the displacement in such a way as to allow for the maintenance of heat pumping against larger temperature differences. Viscous drag is generally a loss or hindrance to any operation. It is quite a surprise to find it here as an asset to a desired object.

SURFACE HEAT-PUMPING CYCLE

The heat-pumping mechanism is complicated by the fact that each tiny mass of gas follows a different cycle. There is a similarity in the general nature of the action but a great difference in the magnitude. The basic action of each little mass of gas is to move back and forth in the tube due to pressure changes. Associated with the pressure change is a temperature change which causes heat to be exchanged with the wall, such that heat received, δQ, near the filling end of the tube, A, as shown in Fig. 3, is returned to the tube at a point further toward the closed end, B, where the tube is at a warmer temperature. Actually, the heat from even a tiny mass of gas will be received and transferred to the tube over a range of positions. However, the average effect will be that of receiving heat from the wall in the vicinity of A and delivering it back to the wall in the vicinity of B.

Different masses of gas will act very differently. Gas very near the walls will not move nearly so far, as shown by A_1 and B_1. It will, however, exchange more heat with the walls for two reasons: it builds up greater temperature differences with respect to the adjacent wall temperature and it is closer to the wall. Gas near the closed end of the closed tube

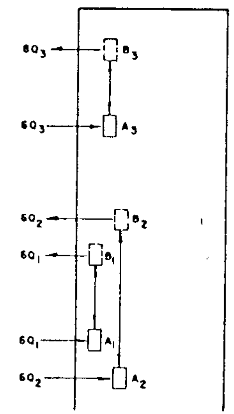


Fig. 3. Heat exchange with the wall.

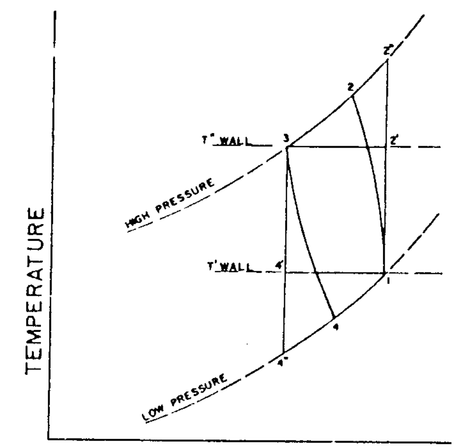


Fig. 4. Surface heat-pumping cycle 1, 2, 3, 4, 1 compared with Carnot cycle 1, 2′, 3, 4′, 1 and Brayton cycle, 1, 2″, 3, 4″, 1.

will also have smaller displacements even at the same distance from the wall, as shown by A_3 and B_3, and it will give a different contribution to the heat pumping. The gas in the center of the tube will contribute very little, if at all.

Though these actions are very different in their magnitude and effective heat transfer, the basic nature of the performance will be very similar for all the effective masses. Its performance may be understood by plotting the cycle for one tiny mass on a T–S diagram, where its cycle may be compared with the Brayton and Carnot cycles (Fig. 4).

Figure 4 shows the cycle followed by a typical mass of gas. Starting from point 1 the pressure is increased and a temperature difference starts to build up, causing a

cooling and a consequent reduction in entropy. As the temperature difference gets larger the cooling increases and results in an increase in the slope of the curve. After attaining the high pressure, which gives the maximum ΔT with the wall at point 2, further cooling occurs if the pressure is held constant. This brings the gas to near equilibrium with the wall at point 3. The expansion occurs with continually increasing heating of the gas by the wall, causing a continual increase in entropy to point 4, followed by additional heating of the gas at constant low pressure from points 4 to 1.

A Carnot cycle, the most efficient possible cycle between the two temperatures, would consist of a reversible adiabatic compression from points 1 to 2′, a reversible isothermal compression from points 2′ to 3, a reversible adiabatic expansion from points 3 to 4′, and a reversible isothermal expansion denoted by points 4′ to 1, completing the cycle. It is interesting to note that the performance approximates rather closely that of the Carnot cycle. Actually, it probably comes closer than any actual working equipment that is being built.

Also shown is the Brayton cycle without regeneration, where all the compression and expansion are reversible adiabatics. This is more characteristic of what might be built into actual equipment, since the achievement of isothermal compression in a reasonably sized piece of equipment running at an acceptable rate of speed has not been accomplished at this time. To remove even half of the heat of compression from helium in a ¾-in.-diameter cylinder would require about ½ sec. Stirling cycle devices operating at 2000 rpm come nowhere near achieving isothermal compression or expansion. In fact, very little heat will be transferred during compression or expansion in the time differential operator ∇^2 is made dimensionless by the diameter squared, D^2, and the thermal diffusivity α^* by a characteristic value of α. Equivalent performance will be obtained in a pulse tube if the value of the characteristic group, α/ND^2, is the same.

This conclusion is borne out by experimental work; ¾-in. tubes give optimum performance at speeds of about 30 to 40 cpm and ⅜-in. tubes perform best at speeds of 120 to 150 cpm with helium gas as the operating fluid. It is not a very distinct optimum speed, since the ⅜-in. tube works quite well from 70 to over 200 cpm

Other aspects of this prediction are also verified by experiments. Air with its lower thermal conductivity requires lower speeds, and the lower densities given by lower pressures lead to lower pulse rates for best performance.

The length does not affect this optimum pulse rate. This prediction indicates that any chamber where it is hoped the Surface Heat Pumping would be a maximum should have a constant equivalent diameter, so that all of it can operate at the ideal pulse rate.

The fact that there is an optimum speed which is quite small for large volumes is partially responsible for its not being noticed before. For example, a 1½-in. tube would have an ideal speed of only about 8 pulses/min, a rather uncommon frequency and heat flow in such a case is very small per unit area. Therefore, not very large temperature differences would be built up to be noticed. Larger chamber would have such small optimum pulse rates and heat flow rates as to make it very unlikely that it would be noticed. There have probably been many unexplained hot spots in equipment involving gas under pressure where fluctuations occur which were due to this phenomenon.

Another interesting fact about the nature of the heat-pumping phenomenon may be deduced from the simple picture of heat transfer to and from the gas. The fraction of the total gas in the tube which is effective in the heat transfer at a given value of α/ND^2 will be independent of tube diameter. Thus the refrigeration produced per cycle Q will be proportional to the diameter squared times the length L,

$$q = CD^2L \tag{5}$$

where C is the proportionality constant.

The net refrigeration effect, Q, will be equal to the product of q and N_0, the optimum pulse rate.

$$Q = qN_0 \tag{6}$$

$$Q = CN_0D^2L \tag{7}$$

If operation is to be at the optimum value of the characteristic number, $C_0 = (\alpha/ND^2)_0$, then

$$Q = C\alpha\left(\frac{L}{C_0}\right) \tag{8}$$

The fact that the optimum cooling is equal in tubes of equal length operating at the same pressures has been confirmed roughly by experiments. Since the material to be cooled is proportional to diameter, small tubes should cool much faster, and they do.

TEST RESULTS

A volume configuration as shown in Fig. 5 has been used to demonstrate the surface heat pumping. This tube differs from those described previously [1,2], by having an open-top heat exchanger rather than the closed design, which resulted in a nearly which is available. These compressions are actually very nearly adiabatic, so that the Brayton cycle is thus achieved.

The Brayton cycle is shown as points 1, 2″, 3, 4″, and 1. It is less efficient than both the Carnot and the Surface Heat Pumping cycles. In the Surface Heat Pumping cycle, substantial cooling may be achieved during compression, so that the isothermal compression is at least approached, and during expansion heating occurs, so that isothermal expansion is also approached. Something between isentropic and isothermal expansion and compression must occur for the Surface Heat Pump cycle to occur, and this gives greater efficiency than that of the Brayton cycle.

The Surface Heat Pumping cycle is not regenerative. Therefore, heat must be pumped in a series of steps from one end of the surface to the other as in Pulse Tube Refrigeration. However, it also allows for the efficient heat pumping of heat leak due to conduction or radiation in the tube at any temperature level at which it occurs.

In a well-built Pulse Tube refrigerator operating near its optimum speed, effectively about 50% of the gas will exchange all its heat with the wall in this very efficient Surface Heat Pumping cycle. The remainder of the gas in the tube will just be compressed and decompressed reversibly. As a consequence it is possible to build up very large temperature differences in small short tubes in a very short time. A ⅜-in.-diameter tube, 7 in. long, being pulsed at 120 pulses/min with helium at 300 psia will frost at one end and be too hot to touch at the other end in two minutes.

Surface Heat Pumping can be a substantial effect. Significant amounts of heat can be pumped against large temperature differences with an efficiency approaching that of the Carnot cycle.

OPTIMUM PULSE RATE

The basic mechanism which produces the heat pumping is initially a pressure change which builds a temperature pattern in the gas followed by heat transfer to the walls. This is followed by a decrease in pressure, building another temperature pattern in the gas followed by heat transfer from the walls to the gas. The temperature patterns are such that heat received from the walls is transferred back to the wall at a point closer to the closed end giving the heat pumping effect.

The pressure changes may be made at any speed. Time, however, must be allowed for heat transfer from the gas to the wall. The time can be either too short, so that very little of the heat is transferred to the walls, or so long that time is wasted when little or nothing is happening. There thus is an optimum pulse rate.[3]

Determining the exact temperature changes in the gas with pressure, displacement, and heat transfer is very complex. However, due to a great similarity in problems of nonsteady heat transfer from volumes, it is possible to deduce some fundamental characteristics about the optimum pulse rate for a round closed tube even though the exact time–temperature pattern in the tube is not known.

If most of the heat is exchanged with the walls at a constant pressure, then the significant terms in the heat transfer equation are

$$\frac{\partial T^*}{\partial \tau^*} = \frac{\alpha}{ND^2}\alpha^*\nabla^2 T^* \tag{4}$$

where the superscript * denotes the parameters which have been made dimensionless by the characteristic group α/ND^2. In this equation, time, τ, is made dimensionless by the reciprocal of the pulse rate N; the incremental distance squared associated with the

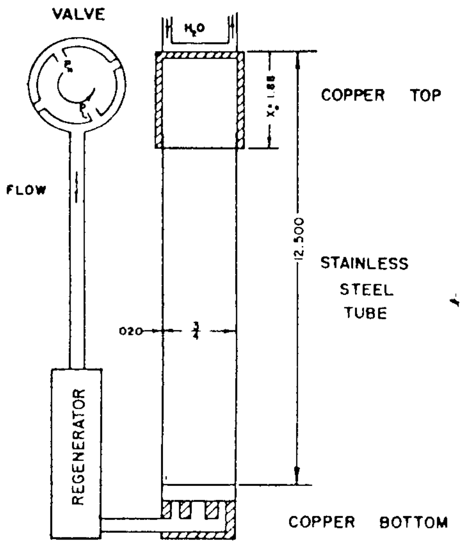


Fig. 5. Pulse tube refrigeration configuration.

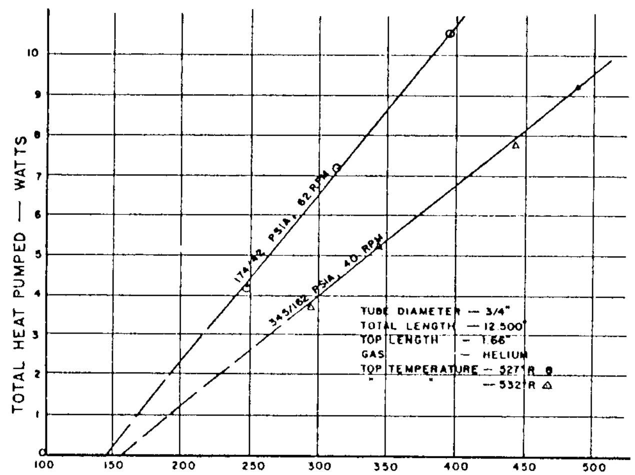


Fig. 6. Total heat pumped *vs.* cold end temperature, test results.

isothermal process at the top. Operating at 62 pulses/min with helium pressures of 174/42 psia, the bottom portion cooled to 247°R while the top remained at 527°R. The heat being pumped was determined by measuring the heat flow out of the top with a calorimeter. This heat flow is plotted *vs.* the cold end temperature in Fig. 6 for the above test and also for one at 40 pulses/min and pressures of 345/162 psia. Projecting the experimental curves down to the condition of no heat being pumped shows that if this condition could be reached the ideal temperature pattern given by (3) would be achieved. The heat being removed at the lowest temperature attainable consisted primarily of the regenerator thermal losses along with a small loss due to radiation.

CONCLUSIONS

A few basic conclusions about Surface Heat Pumping may be summarized here.

1. If any closed chamber is pressurized and depressurized by delivery and exhaustion of gas from one point on its surface and the flow is essentially smooth, heat pumping will occur away from the point on its surface.
2. The Surface Heat Pumping cycle is quite efficient for the gas which participates in a system designed for its utilization. The efficiency for this gas approaches that of the Carnot cycle.
3. The use of this phenomenon in a well-designed system has potential for high efficiency in a cryogenic refrigerator.
4. There is an optimum pulse rate independent of length but dependent on the equivalent diameter of the chamber squared.
5. Although the rate of heat pumping is dependent on the pressures, the minimum achievable cold end temperature for a given gas flow pattern is the same for all pressure ratios and is a function of the volume geometry.

NOTATION

C = constant
D = diameter
L = length
N = pulse rate
P = pressure
Q = quantity of heat
q = refrigeration per pulse
T = temperature
X = position
α = thermal diffusivity
γ = ratio of specific heats
τ = time

Superscripts
$'$ = initial condition
$''$ = final condition
$*$ = dimensionless quantity

REFERENCES

1. W. E. Gifford and R. C. Longsworth, *Trans. ASME, Ser. BJ. Eng. Ind.* (August 1964), p. 264.
2. W. E. Gifford and R. C. Longsworth, in: *International Advances in Cryogenic Engineering*, Plenum Press, New York (1965), p. 69

United States Patent [19]

Bernstein

[11] **3,748,502**

[45] **July 24, 1973**

[54] **PIEZOELECTRIC HELMHOLTZ RESONATOR FOR ENERGY CONVERSION**

[75] Inventor: **Julius Bernstein,** Beechhurst, N.Y.

[73] Assignee: **Edo Corporation,** College Point, N.Y.

[22] Filed: **Aug. 4, 1971**

[21] Appl. No.: **169,111**

Related U.S. Application Data

[63] Continuation of Ser. No. 3,249, Jan. 15, 1970, abandoned.

[52] **U.S. Cl.** **310/8,** 310/8.2, 310/9.6, 310/9.8, 73/194

[51] **Int. Cl.** **H04r 17/00**

[58] **Field of Search** 310/8, 8.1, 8.2–8.7; 417/322

[56] **References Cited**

UNITED STATES PATENTS

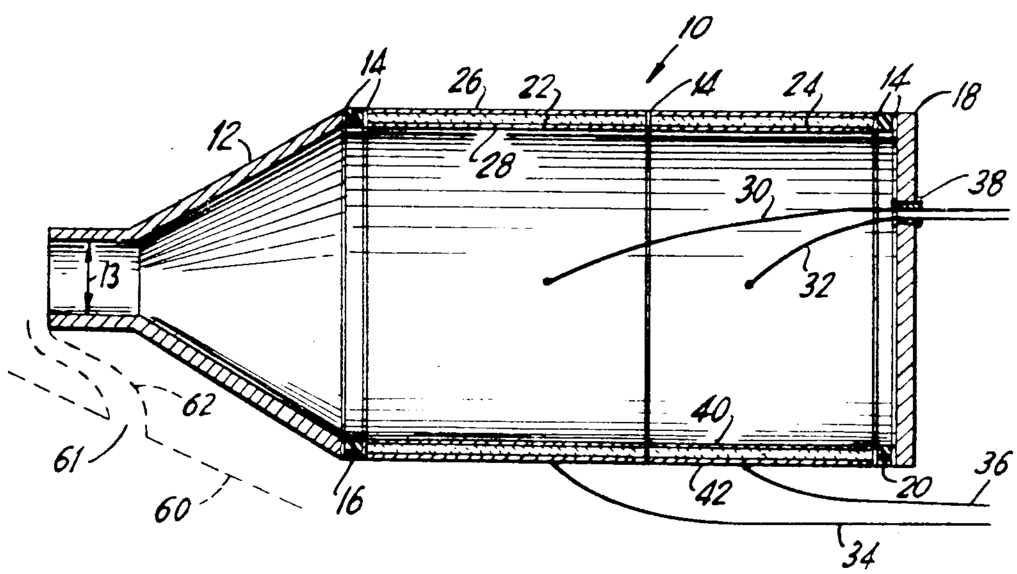
3,150,592	9/1964	Stec	310/8 X
3,215,078	11/1965	Stec	310/8 X

Primary Examiner—J. D. Miller
Assistant Examiner—Mark O. Budd
Attorney—Davis, Hoxie, Faithfull & Hapgood

[57] **ABSTRACT**

A Helmholtz resonator comprising an orifice and an otherwise enclosed volume includes one or more cascaded hollow piezoelectric cylinders. The resonator effects a conversion between an air flow and electrical energy, one energy form being applied as an input to excite the resonator and the other energy form being directly derivable as an output.

9 Claims, 3 Drawing Figures



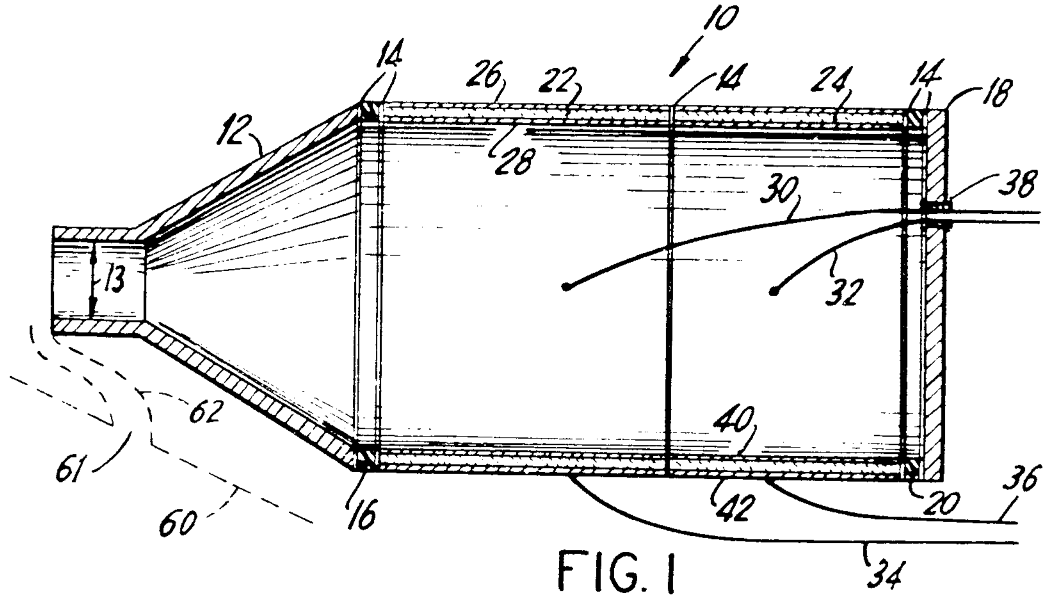


FIG. 1

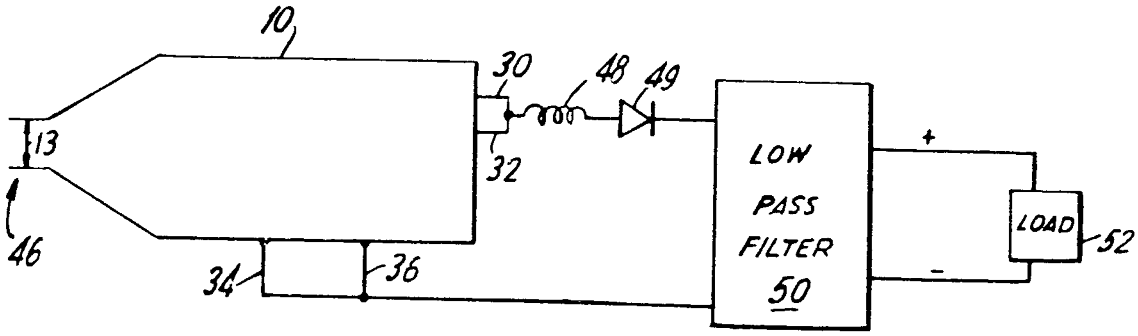


FIG. 2

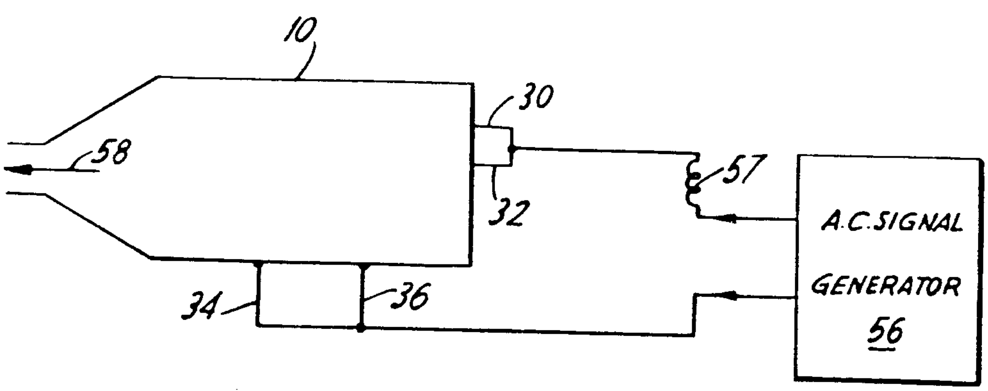


FIG. 3

INVENTOR.
JULIUS BERNSTEIN
BY
Davis, Hoxie, Faithfull & Hapgood
ATTORNEYS

PIEZOELECTRIC HELMHOLTZ RESONATOR FOR ENERGY CONVERSION

RELATED APPLICATIONS

This application is a continuation of Ser. No. 3,249 filed Jan. 15, 1970, now abandoned.

This invention relates to conversion apparatus and, more specifically, to a piezoelectric Helmholtz resonator for converting between an air (or other fluid) flow and electrical energy.

It is desirable for some applications to employ apparatus for directly converting between the kinetic energy of a fluid flow and electrical energy. For example, an electrical voltage source is required to arm and detonate proximity, dela· :d, and similar fusing circuits in artillery projectiles v the physical constraints of the situation, the proje le must contain its own power source and, accordingly, carries batteries to activate the detonator circuitry.

However, there are several deficiencies associated with battery powered projectiles, e.g., the batteries deteriorate with time or severe environmental exposure, and a projectile which includes a battery may inadvertently detonate upon receiving a sharp impact.

In accordance with the present invention, the battery can be eliminated, and the projectile rendered fail-safe, by energizing the projectile arming electronics with a converter which provides an electrical output responsive to the air flow past the fired shell. Similar utility exists for powering varying functional circuits on other vehicles moving in air, or circuits used where an air flow is readily accessible.

Correspondingly, it is often desirable to provide and use a moving air flow, as to cool electrical circuitry. A motor driven blower is typically employed to effect the requisite air flow, but motors often fail in time by reason of various mechanical malfunctions thus causing circuit components to overheat and fail. The problem is especially pronounced where the electronic apparatus is enclosed in a sealed cabinet, or is otherwise inaccessible. The fan is often the least reliable element in the enclosure and thus limits the useful life and/or reliability of the electronics associated therewith.

It is thus an object of the present invention to provide improved energy conversion apparatus.

More specifically, an object of the present invention is the provision of reliable apparatus for converting between electrical energy and the kinetic energy of an air flow.

The above and other objects are realized in illustrative energy converting apparatus which employs a Helmholtz resonator including a piezoelectric structural element, the resonator defining a closed volume having an orifice in one end thereof. A suitable air flow is directed across the orifice and induces a mechanical oscillation within the body of the resonator at a natural frequency thereof, thereby inducing periodic mechanical stress perturbations within the piezoelectric material.

An alternating current potential is developed between two electrodes disposed on opposite surfaces of the piezoelectric material responsive to the periodic stress variation. The alternating current potential may be employed to drive a suitable load, or may be rectified and passed through a low pass filter to energize a load with a direct current potential.

By employing the inverse actuation, i.e., by supplying an alternating current signal corresponding to a resonance frequency of the Helmholtz resonator to the two electrodes, a mechanical resonance is effected. The resonator thereby generates an air flow which exits from the resonator orifice. The air flow may be employed for any desired purpose, e.g., for circuit cooling.

The above and other objects, features and advantages of the present invention are realized in illustrative energy converting apparatus, described herein-below in conjunction with the accompanying drawing, in which:

FIG. **1** illustrates in cross-section a Helmholtz resonator, employing a piezoelectric material, which embodies the principles of the present invention;

FIG. **2** schematically depicts the resonator of FIG. **1** employed to provide output electrical energy responsive to an incident air flow; and

FIG. **3** schematically depicts the resonator of FIG. **1** employed to produce an output air flow responsive to an input electrical energization.

Referring now to FIG. **1**, there is shown a Helmholtz resonator **10** including a first, tapered end section **12** which includes an orifice **13**, and a second end portion **18**, e.g., both formed of aluminum or other material. Disposed intermediate the resonator end sections **12** and **18** are two hollow piezoelectric cylinders **22** and **24**, e.g., formed of barium titanate or lead zirconate. The piezoelectric cylinders **22** and **24** have coated thereon outer silvered electrode surfaces **26** and **42**, and inner silvered electrode surfaces **28** and **40**, respectively. Any number of piezoelectric cylinders (or only one) may be cascaded together within the resonator **10** for increased energy converting capacity.

Two spacers **16** and **20** separate the piezoelectric material from the end resonator sections **12** and **18**, the spacers preferably being of a resilient insulating material such as rubber, an elastomer, a cork-like material or the like. The several resonator members are secured together by an insulating cement **14** such as an epoxy resin.

The insulating spacers **16** and **20**, and the several cement joints **14**, do not short circuit the electrode surfaces **26, 28, 40** and **42**, and, moreover, inhibit the metallic end members **12** and **18** from spuriously interconnecting these conductors. The inner and outer electrode surfaces must be electrically isolated or the converter is disabled — all four surfaces must be isolated for serial electrical operation. The resilient nature of the spacers mechanically "floats" the piezoelectric cylinders **22** and **24** relative to the end members **12** and **18**, i.e., allows the members **22** and **24** to expand and contract, and to thereby provide energy conversion in a relatively efficient manner as discussed below.

Finally, conductors **30, 34, 32** and **36** are connected to the electrode surfaces **28, 26, 40**, and **42**, respectively, the inner conductors **30** and **32**, insulated from each other, passing through a seal **38** (a grommet or the like) about an aperture within the resonator member **18**. The wires **30** and **32**, and **34** and **36** may be interconnected to provide parallel electrical operation of the cylinders **22** and **24** (corresponding to a relatively low impedance, low voltage and a relatively high current output or input), or the wires **30** and **36**, or **34** and **32** can be connected (relatively high impedance, high voltage, low current output or input) to effect series cylinder operation.

As is well known, the Helmholtz resonator of FIG. 1 has a plurality of natural mechanically resonant frequencies when excited, the different frequencies corresponding to differing vibrational modes (e.g., sinusoidally varying axial length, diameter and the like). The specific frequency values depend upon the mechanical resonator configuration, e.g., its dimensions, thickness, bulk material properties and the like.

When air of a suitable velocity and incident angle is directed across the orifice **13**, the resonator **10** will resonate in a corresponding mode thereby impressing periodic sinusoidal stress variations in the cylinders **22** and **24**. By reason of their piezoelectric properties, the cylinders **22** and **24** induce like phased bipolar sinusoidal voltages between the conducting electrode surfaces **26** and **28**, and **42** and **40**, and thereby also between the wires **34** and **30**, and **36** and **32**, respectively. These alternating current potentials may be employed per se to drive the same or different loads, or may be connected in series or parallel as described above to increase the voltage or current output of the resonator **10**.

Alternatively, a sinusoidal electric potential corresponding to selected of the natural frequencies of the resonator **10** (or a periodic waveform having a Fourier component at such a frequency) may be applied via the conductors **30–34** and **32–36** to the electrodes **28–26** and **40–42**. Accordingly, stresses are produced in the piezoelectric cylinders **22** and **24** which cause the unit **10** to resonate at the applied frequency. The vibration of the resonator walls at selected ones of the resonator natural frequencies causes an air movement outward through the resonator orifice **13**.

An air (or other fluid) flow-to-electrical output converter employing the resonator **10** is schematically shown in FIG. **2**, the output electrodes **30–32** and **34–36** being connected for parallel operation for convenience. An air flow **46** past the resonator aperture **13** induces an alternating current potential between the output electrodes as discussed above. The flow **46** may be provided by a compressed air source, or by motion of the resonator **10** relative to a surrounding air environment. Thus, for example, the resonator may be located inside an artillery projectile with the orifice **13** comprising an opening on the surface of the shell (or such an opening connected by a conduit to direct air past an internally located orifice **13**). FIG. **1**, for example, shows in dotted line form the surface of an artillery projectile **60** having opening **61** connected by conduit **62** to direct air flowing past said opening past the orifice **13**.

The alternating potential may be utilized to energize a load directly, or via a series inductor. Alternatively, the potential may be converted to direct current form as by a rectifier **49** and a low pass filter **50**.

A series inductance **48** is employed to develop a reactance at the operative resonance frequency which offsets, or at least partially cancells the capacitive reactance exhibited by resonator **10**. The effective electrical output of the resonator **10** is thus converted from a high reactive impedance to a low impedance of small phas angle, hence greatly increasing the power supplied by the resonator **10** to a low impedance load.

The potential developed by the resonator **10** connected as in FIG. **2** may drive any suitable load, such as a projectile detonator circuit. When employed in an artillery projectile application, the unipolar potential generating apparatus of FIG. **2** is fail-safe, the threshold air flow required for detonation being made sufficiently great as to be unobtainable until the projectile has actually been fired and has left the firing weapon. Further, the apparatus can be stored indefinitely, and under adverse conditions, without loss of potency.

An electrical eneryg-to-air flow converter is schematically shown in FIG. **3**, and includes a signal generator **56** driving the resonator with a sinusoidal signal (or signal component) corresponding to a resonator resonance frequency. A series inductor **57** may advantageously be employed to offset the large capacitive reactance of the resonator **10** thereby increasing its energy transfer efficiency in a manner directly analogous to that described above for the arrangement of FIG. **2**. As discussed above, the resonator **10** responds to the input energization by generating an outward output air flow **58** through the orifice **13**. For the above-described cooling application, the air flow may be directed at a heat producing electrical component to provide circulation thereabout, or may be directed along the surface of a metallic enclosure to increase heat transfer through the enclosure.

Thus, the Helmholtz resonator **10** of FIG. **1** has been shown by the above to effect a direct conversion between energy forms. The conversion apparatus is free of any moving parts in the conventional sense, and exhibits a reliability comparable to the solid state components with which it may often be associated.

The above-described resonator construction and ancillary apparatus is merely illustrative of the principles of the present invention. Numerous modifications and adaptations thereof will be readily apparent to those skilled in the art without departing from the spirit and scope of the present invention. For example, the active piezoelectric cylinders **22** and **24** can be formed of the well known "striped" cylinder or ring construction (k_{33} mode vis.-a-vis. the k_{31} mode considered above).

What is claimed is:

1. A Helmholtz resonator for converting between kinetic fluid energy and electrical energy, comprising first and second resonator end means and a Helmholtz resonator cavity therebetween, a hollow piezoelectric element disposed between said resonator end means and defining the side enclosure of said resonator cavity, first and second electrode means located respectively on the outer and inner surfaces of said hollow piezoelectric element, conductors connected to said electrodes, said first end means having an orifice therein, said second end means defining a cavity closure, and means for directing a fluid flow across said orifice, such that fluid flow across said orifice induces oscillations within the resonator cavity to induce periodic stress variations in the piezoelectric element and a voltage across said electrodes.

2. A combination as in claim **1** for arming an artillery projectile, further comprising an aperture located on the surface of said projectile such that a portion of the air flow past said aperture flows across said orifice, and circuitry connected to said conductors, said circuitry being activated when a voltage is induced across said conductors.

3. A combination as in claim **1,** further comprising resilient insulating means spacing said piezoelectric element from said first and second end means.

4. A combination as in claim **1,** wherein the orifice in said first end means has a cross-sectional area con-

stricted from the cross-sectional area of said hollow piezoelectric element.

5. A combination as in claim **1** further comprising an electrical load connected to said conductors.

6. A combination as in claim **5** further comprising a rectifier and a low pass filter connected between one of said conductors and said load.

7. A combination as in claim **1** further comprising an inductance connected to one of said conductors.

8. A combination as in claim **7** wherein said resonator comprises a capacitive reactance at a natural resonance frequency, and wherein said inductance exhibits a substantially like absolute magnitude reactance at said frequency.

9. A combination as in claim **1** wherein said piezoelectric member comprises a plurality of hollow piezoelectric segments adhered together.

* * * * *

United States Patent [19]

Ceperley

[11] **4,114,380**

[45] **Sep. 19, 1978**

[54] **TRAVELING WAVE HEAT ENGINE**

[76] Inventor: **Peter Hutson Ceperley**, 4803 Clemons Ct., Annandale, Va. 22003

[21] Appl. No.: **774,063**

[22] Filed: **Mar. 3, 1977**

[51] **Int. Cl.²** **F03G 7/00**

[52] **U.S. Cl.** .... **60/721**; 62/467 R

[58] **Field of Search** 60/516, 530, 643, 650, 60/682, 721 16/DIG. 22; 62/6, 118, 467

[56] **Refer :es Cited**

U.S. PATENT DOCUMENTS

3,339,635 9/1967 Brandon 62/467

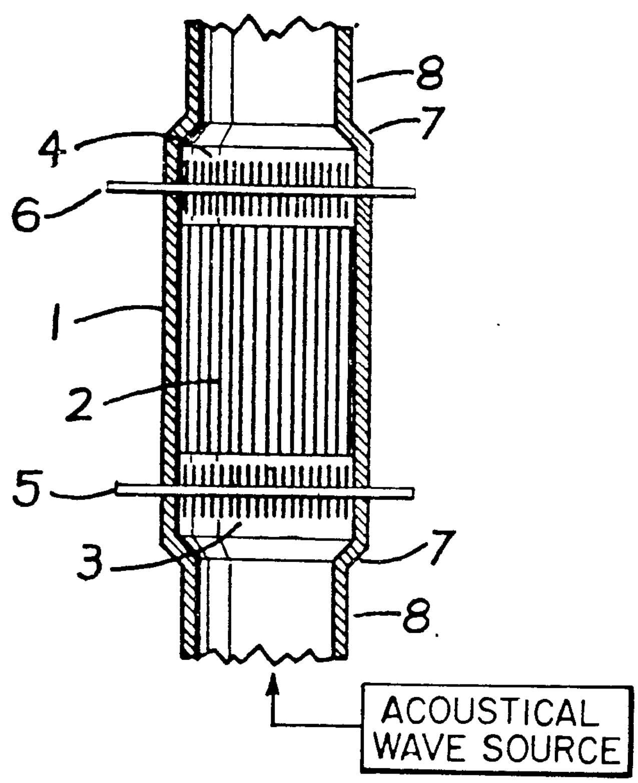
Primary Examiner—Allen M. Ostrager

Assistant Examiner—Stephen F. Husar

[57] **ABSTRACT**

A traveling wave heat engine can be used both as a heat engine and a heat pump. When it operates as an engine, thermal energy is converted into acoustical traveling waves from an acoustical wave source, which if desired, can be further converted into more conventional forms of electrical and mechanical power. When it operates as a heat pump, acoustical traveling waves supply the motivating force to pump heat. An object such as a flame holding screen, heat exchanger, or regenerator creates a stationary temperature gradient in the compressible working fluid of the engine. As acoustical traveling waves pass through this temperature gradient, they cause the fluid there to move through the gradient and back and so be heated or cooled in conjunction with the compression and expansion phases of the wave. The pressure and velocity components of an acoustical traveling wave are inherently phased properly to cause the fluid in the stationary temperature gradient to undergo a Stirling thermodynamic cycle that results in amplification or attenuation of the wave, depending on the wave direction relative to the direction of the gradient. This cycle also pumps heat in the direction opposite the direction of wave propogation through the device.

7 Claims, 7 Drawing Figures



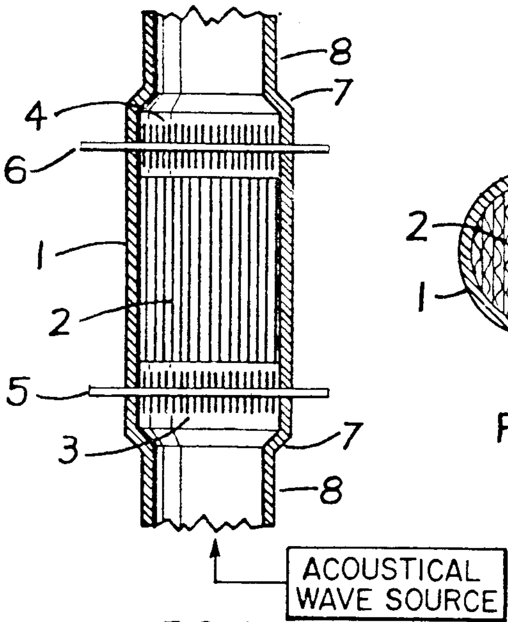


FIG. 1

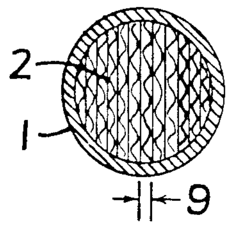


FIG. 2

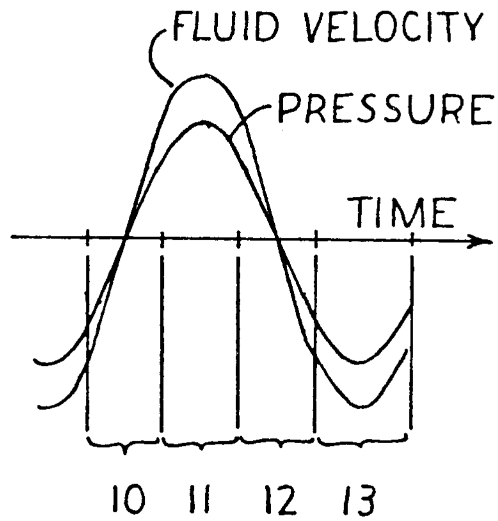


FIG. 3

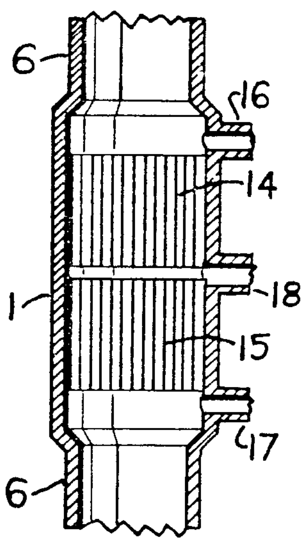


FIG. 4

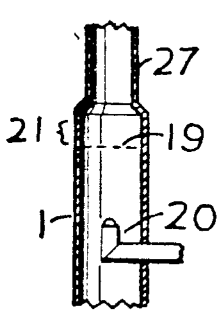


FIG. 5

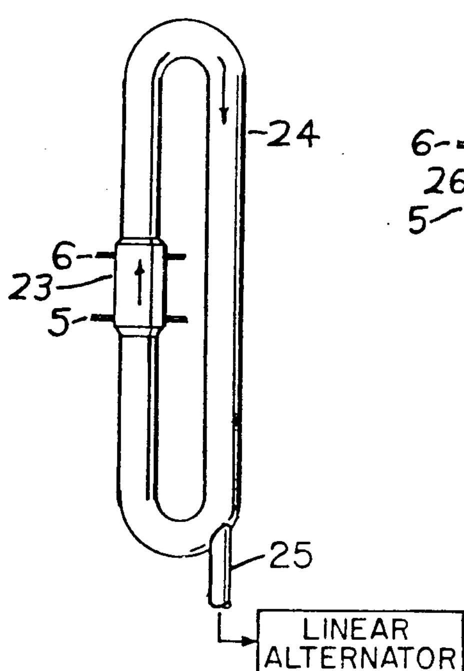


FIG. 6

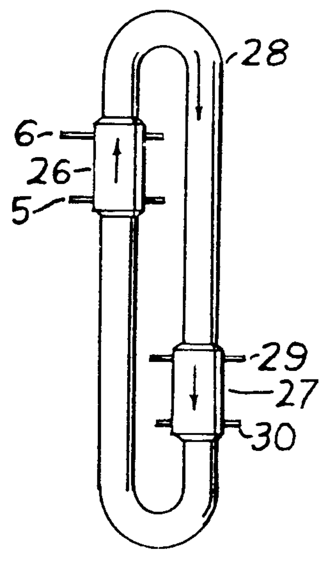


FIG. 7

TRAVELING WAVE HEAT ENGINE

BACKGROUND OF THE INVENTION

This invention relates to a heat engine which uses acoustical traveling waves to cause expansion and contraction of a unit mass of motivating fluid.

It is well known that acoustical standing waves of considerable amplitude can be set up in a gas column by the application of heat at a suitable fixed point. The transformation of heat into acoustical standing waves is accomplished by the action of these standing waves in compressing, expanding, and moving the gas similar to that which is done in a more conventional heat engine by the pistons. The standing waves simultaneously compress and move the fluid through the heat source then expand it while moving it in the reverse direction. For best operation, the heat exchange rate between heat source and the gas should be adjusted to provide a delay time of approximately one quarter of an acoustical period, so that on the average, the fluid is heated after the compression has taken place, but before the expansion. While standing wave heat engines have great appeal because of their lack of moving parts, the need for a thermal delay prohibits use of very good heat exchange and subsequent attainment of high efficiencies.

CROSS REFERENCES

Part of this invention was disclosed in Disclosure Document No. 051914, "Traveling Wave Heat Engine", received by the U.S. Patent and Trademark Office on Aug. 23, 1976.

SUMMARY

This invention uses traveling waves to accomplish the compression, expansion, and gas movement through the heat source, instead of the above mentioned standing waves. Because traveling waves are phased differently than standing waves and have their fluid movement timed to occur between the compression and expansion phases, in a traveling wave heat engine no thermal delay need be provided and efficient heat exchange can be used. This allows higher efficiencies to be attained.

It is the object of this invention to provide an improved acoustical heat engine for the efficient transformation of thermal energy into acoustical energy and vice versa.

Another object of this invention is to provide a new source of mechanical and electrical power.

Another object of this invention is to provide an efficient, thermally driven heat pump, with no major moving parts.

And yet another object of this invention is to provide such an acoustical heat engine which uses a regenerator and the Stirling thermodynamic cycle.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a cross sectional view of a simple traveling wave heat engine using a hot and a cold heat exchanger, and a regenerator to produce a spacially fixed temperature gradient.

FIG. **2** is another cross sectional view of the heat engine in FIG. **1**, showing the end view of the regenerator packing.

FIG. **3** is a graph of the pressure and velocity versus time for a small fluid volume through which an acoustical traveling wave is passing.

FIG. **4** is a cross sectional view of a variant of FIG. **1**, using hot and cold fluid flows to add and extract heat to and from the engine.

FIG. **5** is a cross sectional view of a variant of FIG. **1**, using combustion of fuel and a flame holding screen to set up a stationary temperature gradient.

FIG. **6** is a side view of an acoustical oscillator, using the device of FIG. **1** as the gain element.

FIG. **7** is a side view of a thermally driven heat pump, using two of the devices of FIG. **1**.

DESCRIPTION OF THE PREFERRED EMBODIMENT

FIG. **1** shows the preferred embodiment of a traveling wave heat engine. It consists of a casing **(1)**, a regenerator **(2)**, and two heat exchangers **(3)** and **(4)**. FIG. **2** shows an end view of the preferred regenerator packing **(2)**, which consists of thin, corrugated stainless steel sheets interleaved with thin, flat stainless steel sheets. Other standard regenerator packing might also work, however best operation might be expected from those packings which have been developed for Stirling engine regenerators. To convert thermal energy into acoustical energy, acoustical traveling waves are made to enter through the bottom of the heat engine, while at the same time the bottom heat exchanger **(3)** is cooled by passing externally chilled liquid through pipe **(5)**, and the top heat exchanger **(4)** is heated by passing externally heated liquid through pipe **(6)**. The heated and cooled heat exchangers set up a continuous temperature gradient in the regenerator **(2)**. The regenerator packing is chosen fine enough so that the working fluid in the regenerator is essentially in thermal equilibrium with the packing around it, but not so fine as to prevent the passage of acoustical waves through the regenerator. The upwardly moving traveling waves first pass through the cold heat exchanger (3), are amplified in the regenerator (2), and exit through the hot heat exchanger **(4)**.

The amplifying ability of the regenerator can be understood by considering the action of a small volume of fluid in the regenerator. If a wave induces the fluid to move up towards the hot end, the fluid is heated by the hotter regenerator packing there and expands; if it induces the fluid to move down towards the cold end, it is cooled and contracts. As acoustical traveling waves pass through any compressible media, the pressure and velocity they impart to a small volume of the media is shown in FIG. 3 as a function of time. As shown, traveling waves are intrinsically phased such that the peak velocity and the peak pressure occur at the same time. The phases that a small volume of the fluid in the regenerator undergoes as the waves travel through it are: first a build up of pressure and compression **(10)**, then a flow of fluid towards the hot end and heating **(11)**, then a drop in pressure and expansion **(12)**, and finally a flow of fluid towards the cool end and cooling **(13)**. This is the basic thermodynamic cycle of a Stirling engine and results in conversion of thermal energy into mechanical energy. In the case of a conventional Stirling engine, pistons are responsible for compression, expansion, and movement of the fluid, and the resulting energy is transferred to these pistons. In the case at hand, traveling waves cause the compression, expansion, and fluid movement and the energy goes into adding to the pres-

sure and the momentum of these waves, thereby amplifying them. Another way to view this process is that in traveling from cold to hot, the traveling waves move the fluid particles in the compression parts of the waves upwards, heating them and increasing the pressure in these parts. Those in the rarefied parts of the waves are moved downwards and cooled, reducing the pressure in these places. Thus the pressurized and rarefied parts of the waves are further pressurized and rarefied, making the waves more intense. Note that it is not necessary for any single fluid particle to traverse the whole length of the regenerator, only that the waves cause it to move, even slightly, relative to the regenerator and thus cause it to be heated and cooled.

For best operation, the regenerator packing should be fine enough so that any thermal delay is less than one quarter of the period of the traveling wave. At the same time, too fine of packing or too long a regenerator can result in excessive acoustical attenuation. For an air filled, one atmosphere pressure device of the type shown in FIGS. **1** and **2** amplifying 10 Hz waves, the regenerator spacing (**9**) could be about 1 mm. In such case it is important to maintain at least a 1° C/mm temperature gradient along the regenerator so that the acoustical gain is greater than the attenuation due to the viscous losses.

Also, it is important that the acoustical reflections be minimized. This can be accomplished by tailoring the dimensions of the enclosing pipe and various parts to keep the characteristic impedance constant constant along the length of the engine, analogous to that which is done in microwave components and circuits. FIG. **1** shows an increase in the diameter of the engine casing (**1**) over that of the connecting pipes (**8**) which compensates for the presence of the heat exchangers, the regenerator, and the temperature gradient inside the engine. Alternately, an adjustable iris or projection might be used to null out reflections caused by these components, by creating oppositely phased reflections which destructively interfere with the first reflections.

As drawn, the device is vertical with the hot end at the top so that the hot fluid floats on top of the cold fluid and so that convection, which would waste energy, does not occur. Membranes or flexible baffles which would allow the passage of sound, but not of steady fluid flow, might also be used to prevent convection, independent of the device orientation. Obviously, these engines should be well insulated to prevent thermal conduction losses.

As with many other heat engines, the traveling wave heat engine can be driven in reverse and be used as a heat pump, to convert acoustical energy into thermal energy. In this case, the traveling waves are made to go from hot to cold through the stationary gradient, so that the wave induced fluid velocity is in the opposite direction as before and the thermal cycle is now: compression, cooling, expansion, and heating. This results in the attenuation of the sound waves and building up of the thermal gradient.

For example, the device in FIG. **1** could be used to produce cold, for refrigeration, air conditioning, etc., by moving the acoustical wave source to the top of the heat engine and making the traveling waves propagate downwards through the device while ambient temperature coolant is passed through the top heat exchanger pipe (**6**) to remove the heat pumped there. The traveling waves going through the regenerator will automatically pump heat from the bottom end of the regenerator up to the top end. The cold generated at the bottom of the regenerator can be extracted by passing coolant through pipe (**5**) which will be chilled inside the heat exchanger (**3**). This chilled coolant can then be used as desired. Conversely, if ambient temperature coolant is passed through pipe (**5**), heat will be pumped to and concentrated at the top of the regenerator and could be extracted through pipe (**6**) for use elsewhere.

Note that in the above discussion, the waves pump the heat in a direction opposite to their propagation. Actually, these waves determine the direction and, also, the rate of heat flow through the device. Note, that even in the case of the traveling wave heat engine (acting as a heat engine rather than a heat pump) the heat flow direction is still opposite the wave propagation direction. This action is analogous to that occurring in a positive displacement water pump: the pump's rotational speed and direction determine the rate and direction of the water flow through it. In the case of the pump, the pressure of the water on the input as compared with that on the output determines whether the pump "pumps" the water or the water "drives" the pump. Similarly, in the case of the present invention, the temperature of the thermal input as compared with the output determines the direction of energy transformation between acoustical and thermal forms. If the input is hotter than the output (as it is in the case of a traveling wave heat engine), the "thermal potential" "drives" or amplifies the waves. On the other hand, if the input is colder than the output (as it is in this device used as a heat pump), the "reverse thermal potential" "bucks" or attenuates the waves and draws energy from them, as is necessary for pumping heat.

ALTERNATIVE CONSTRUCTIONS

The device as outlined in the claims requires means to add and to extract thermal energy from regions containing the fluid, a stationary object which causes the temperature gradient between the regions to be stationary, and a means for causing traveling waves to propagate through the device along the desired path through the device. In the preferred embodiment described above, the heat exchangers (**3**) and (**4**) comprise the means to add and extract thermal energy, the regenerator (**2**) comprises the stationary object, and the casing (**1**) and connecting pipes (**8**) comprise the means for directing the wave propagation. There are many ways, using conventional devices, in which one can depart from the preferred embodiment: in the means for adding and removing heat, in the object for making the temperature gradient stationary, and in the means for directing the acoustical traveling waves. I shall mention a few of these alternatives here for the sake of illustrating the possible diversity.

FIG. **4** shows one end (**14**) of the regenerator being heated by a flow of hot fluid and the other end (**15**) being cooled by a flow of cool fluid directly over the surfaces of the regenerator. These steady flows are superimposed on the wave induced fluid motion. The hot fluid enters through port (**16**), while the cold fluid enters through port (**17**). These flows pass through the ends of the regenerator, mix in the center, and are exhausted out port (**18**). The input ports (**16**) and (**17**) and the exhaust port (**18**) could be tuned to reduce the loss of acoustical power, escaping out the ports. To use this unit as a heat pump, one would reverse both steady flows of fluid and use the top port (**16**) to extract the cold produced by the heat pumping action of the travel-

ing waves, while using port (17) to exhaust the waste heat.

FIG. 5 shows the temperature gradient set up by a burning zone of fuel and air (21) which is spacially fixed by a screen type flame holder (19). A steady flow of the air-fuel mixture is superimposed on the motion of the traveling wave, to keep the burning zone supplied with energy, as well as cooling the bottom side of the flame screen. Note that in this embodiment, the rate of heat transfer is not temporally uniform as in the other embodiments discussed, but varies in time due to the action of the waves in modulating the flow of air and fuel to the burning zone.

Another possible method for adding heat would be to inject steam or hot water (or any second type of heated fluid) into the hot end of an air filled regenerator (this could also be filled with some other gas) like the type in FIG. 4, and allow the steam or water vapor to condense on and cover the regenerator surfaces. Since the heat of vaporization would be involved in the thermodynamic cycle and this is larger than the normal heat capacity of plain air, such engine might have a higher specific power output than a purely air filled engine.

U.S. Pat. Nos. 2,549,464 and 2,836,033 show acoustical heat engines which are similar to the present invention with the exception that they operate on standing waves instead of traveling waves. In these engines, heat exchangers (and in a few cases, a flame holding screen) serve the dual role of transferring heat and spacially fixing the temperature gradient. In U.S. Pat. No. 2,549,464 the fixing is accomplished by a steady flow of air superimposed on the wave induced air motion, similar to that in the device of FIG. 5 of the present invention. This type of arrangement "sweeps" the temperature gradient up next to the heat exchanger and thus fixes the gradient's location. In U.S. Pat. No. 2,836,033 the fixing is accomplished by locating the hot and cold heat exchangers extremely close to each other, so that the temperature gradient, which must be between them, can move very little, and any substantial fluid motion will cause the required heating or cooling of the fluid. All these methods, both of transferring heat and of fixing the temperature gradient, could easily be used in the present invention. In fact, FIG. 5 uses some of them.

The present invention, as well as the patents referred to above, use pipes to guide and direct the acoustical waves. Alternately, one could use acoustical mirrors, lenses, or other similar devices to focus or define the path of the waves. One might also use an acoustical wave guide which is made of fluid of a different characteristic impedance than that of the surrounding material, analogous to an optical fiber or dielectric wave guide.

APPLICATIONS

The acoustical amplifiers and heat pumps of FIGS. 1, 2, 4, and 5 could be used as general purpose amplifiers and heat pumps in acoustical circuits. FIGS. 6 and 7 are two possible circuits. There could obviously be many analogies drawn between such acoustical circuits and microwave electronic circuits. For instance, the characteristic impedance of the acoustical circuits and components could be tailored much the same as it is in microwave circuits: by controlling the dimensions and properties of the wave guiding channels and media. As in the microwave case, part of an acoustical circuit could consist of lumped-parameter transmission lines or transformers, which might consist of a series of free pistons or a series of masses and springs. Likewise resonant transformers might be used to match impedances of select frequencies, analogous to their use in microwave circuits.

For some applications, it may be desirable to send the same traveling wave more than once through the device in order to increase the energy transfer between the thermal and acoustical forms. FIG. 6 shows one such application, where the traveling wave, after having propagated once through the traveling wave heat engine (23) (which in this case is meant to be the same as that shown in FIG. 1) is returned to the input again by the sound return pipe (24), which is a simple pipe without any packing. Each time the wave travels from cold to hot through the regenerator, it is further amplified. If the amplitude and phase of the returning wave is correct, the device will self-oscillate, generating its own acoustical traveling wave without any other acoustical input and this would comprise an acoustical oscillator. This oscillator is analogous to an electrical oscillator where the sound return pipe provides the positive feedback and the regenerator is the gain element. The temperature gradient in the sound return pipe is not spacially fixed, and moves with the wave induced fluid motion, produces no instantaneous heating or cooling of the fluid particles, and so has little effect on the intensity of the wave.

The sound produced by this acoustical oscillator would mostly consist of a wavelength equal to the distance around the acoustical circuit, although other harmonics might also be present. An acoustical filter could be placed in the circuit to define and control the frequency more exactly. A relief valve or flap might be used to control the amplitude of the sound produced.

The output pipe (25) is used to extract acoustical power for outside use. The sound coming out could be used directly as a foghorn, perhaps by simply adding a flare to the pipe's end, as is done to the pipe end in FIG. 1 of U.S. Pat. No. 2,836,033. It could also be used as a source of pressure fluctuations to drive a piston, a pump, or a special type of turbine designed to be driven by fluctuating pressure. A linear alternator as shown in FIG. 6 might be used to convert this fluctuating pressure into electrical power. Alternately, a steady pressure might be produced by passing these sound waves through a one-way valve, such as a Worthington feather valve, or through a set of such valves arranged to imitate an electrical full wave rectifier. The steady pressure produced there might be used directly to drive a conventional turbine or some other type of expansion engine, or be converted to some other form of mechanical or electrical power. Various other ways of converting the acoustical power output into electrical energy are shown in U.S. Pat. Nos. 2,549,464 and 2,836,033. Most of the figures of these patents show the acoustical waves driving diaphragms or bellows which in turn form or are connected to the moving element of a linear, electrical alternator. One of the figures shows the waves driving a piezoelectric element which produces the electricity. Obviously, traveling wave heat engines, such as the acoustical oscillator in FIG. 6 of the present invention could be used, equally well, to supply acoustical energy to these electrical generating devices.

FIG. 7 shows another acoustical circuit utilizing two traveling wave heat engines. Here, one such heat engine (26) is used to produce sound which drives another one (27) in reverse, which can be used to pump heat. The temperature gradient in the heat engine (26) is produced

by the application of external heat and cold in the heat exchanger pipes (6) and (5), respectively. The top end of the heat pump regenerator is kept at ambient temperature by the flow of coolant through the heat exchanger pipe there (29), while the bottom end of this regenerator is cooled by the action of the traveling waves pumping heat to the top end. The cold produced at the bottom is extracted by the coolant in the heat exchanger pipe there (30). Alternately, one or more control valves and a bypass sound return pipe might be used to vary the fraction of the acoustical power from the heat engine that is sent through the traveling wave heat pump.

I claim:

1. A heat engine comprising:
   a. a compressible fluid capable of supporting propagation of an acoustical traveling wave,
   b. a means for adding thermal energy to the fluid in one region of space (region A),
   c. a means for extracting thermal energy from the fluid in another region of space (region B),
   d. at least one stationary object in and/or between the two regions, in approximate thermal equilibrium with the fluid there, for the purpose of causing the temperature gradient which can exist in the fluid there to remain essentially stationary in the presence of wave induced fluid motion through the gradient, and
   e. a means for causing an acoustical traveling wave to propagate on a path through the fluid from region B, through the region of stationary temperature gradient, into region A, in such a manner that the wave moves fluid back and forth through the stationary temperature gradient and so causes the fluid there to be heated and/or cooled in conjunction with the compression and expansion of the wave, for the purpose of transforming energy between the thermal form of the temperature gradient and the acoustical form of the traveling wave.

2. The heat engine in claim 1 in which region A is maintained at a higher temperature than region B, such that the traveling wave propagates from cold to hot through the stationary temperature gradient, for the purpose of transforming thermal energy into acoustical energy.

3. The heat engine of claim 2 further characterized by a means for converting the acoustical energy produced by said heat engine into other forms of electrical and mechanical energy.

4. The heat engine of claim 2 further characterized by a means for causing part of the traveling wave to repeat its passage through the stationary temperature gradient in the original direction, this part being of sufficient intensity and proper phase that the acoustical circuit will self-oscillate and produce acoustical power without an acoustical input.

5. The heat engine of claim 1 in which region A is maintained at a lower temperature than that of region B by the heat pumping action of the traveling waves through the stationary temperature gradient, for the purpose of providing a source of heat or cold.

6. The heat engine of claim 1 in which the means for causing the temperature to remain essentially stationary comprises a regenerator.

7. The heat engine of claim 6 in which the means for adding thermal energy and the means for extracting thermal energy comprise flows of hot and cold fluid, respectively, through parts of the regenerator.

* * * * *

Journal of the Acoustical Society of America

A pistonless Stirling engine—The traveling wave heat engine

Vol. 66, No. 5
November 1979
p. 1508-13.

Peter H. Ceperley

Department of Physics, George Mason University, Fairfax, Virginia 22030
(Received 11 April 1979; accepted for publication 9 May 1979)

The propagation of acoustical waves through a differentially heated regenerator results in gas in the regenerator undergoing a Stirling thermodynamic cycle. One direction of wave propagation results in amplification of the waves and conversion of thermal energy into acoustical energy. The opposite direction results in acoustical energy being used to pump heat. The ideal gain and maximum energy conversion rates are derived in this paper. Low power gain measurements were made which verify the derived gain equation. Practical engines and heat pumps using this principle are discussed.

PACS numbers: 43.28.Kt, 43.88.Ar

INTRODUCTION

Unfortunately, most machines capable of converting thermal energy into mechanical energy are quite complex. However, if one accepts acoustical energy as being a fluctuating form of mechanical energy, there is a class of very simple devices, called singing pipes (shown in Fig. 1), capable of such conversion. These use standing waves in pipes to force gas to undergo a cycle of compression, heating, expansion, and cooling similar to that of normal heat engines, and similarly convert heat into mechanical energy. In a singing pipe, the heat is normally supplied by a flame heating an object in the resonant tube and the mechanical energy produced goes into maintaining the standing wave. Singing pipes are usually thought of as demonstration devices, serving no useful purpose other than to make a loud noise. However in 1948 and 1952 Bell Telephone Laboratories received patents[1,2] on singing pipes coupled to acoustical-to-electrical transducers which could produce useful electrical power. While these devices were attractive because of their simplicity, they were inefficient because of their use of standing waves. This inefficiency is explained later.

In this paper, we shall explore a different, but similar class of acoustical heat engines, based on traveling waves, which promise higher efficiencies. These traveling wave heat engines[3] use a Stirling thermodynamic cycle, which is reversible, allowing the engines to also serve as heat pumps.

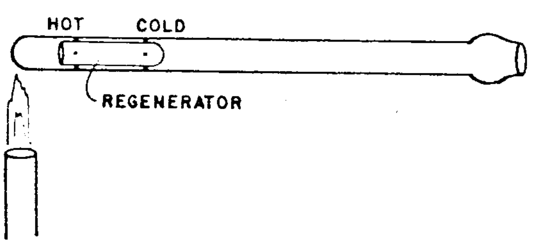


FIG. 1. A singing pipe. Sound is produced when the closed end is placed in a flame. This device uses the acoustical standing waves set up in the pipe to force the gas there to undergo a cycle of compression, heating, expansion, and cooling, similar to that in a normal heat engine. In this case the thermal energy is converted into acoustical energy which maintains the standing waves.

I. THE ENERGY CONVERSION PROCESS

The basic energy conversion process involves an acoustical traveling wave propagating through a differentially heated regenerator, as shown in Fig. 2. The regenerator consists of a casing packed with metal or ceramic parts, small enough to insure that gas in any part of the regenerator is essentially at the temperature of the packing at that point, but not so fine as to cause excessive attenuation of the acoustical waves. This trade-off limits the power density as discussed below. A continuous temperature gradient is set up along the length of the regenerator by external sources which heat one end and cool the other. When a wave forces a volume of gas to move towards the hot end, it is heated by the hotter regenerator there, when it is moved towards the colder end, it is likewise cooled.

The pressure and velocity that a traveling wave imparts to the gas volume it is propagating through is shown in Fig. 3 as a function of time. For a wave traveling from cold to hot through a differentially heated regenerator, this would cause (1) a build-up of pressure (compression), (2) then a flow of gas towards the hot end (heating), (3) followed by a drop in pressure (expansion), and (4) finally a flow of gas towards the cool end (cooling). Since this is the same type of cycle a gas volume would undergo in a standard Stirling engine,[4,5] one would expect a similar conversion of thermal energy into mechanical energy. However, since the acoustical wave is responsible for the compression, expansion, and gas movement, the mechanical energy produced by the cycle in the traveling wave heat engine will amplify the wave.

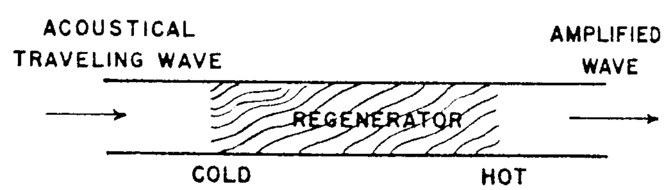


FIG. 2. Traveling wave heat engine. This device is similar to a singing pipe except in its use of traveling waves instead of standing waves. Acoustical traveling waves propagating through the differentially heated regenerator from cold to hot are amplified. This device is reversible and promises higher efficiency than the singing pipe.

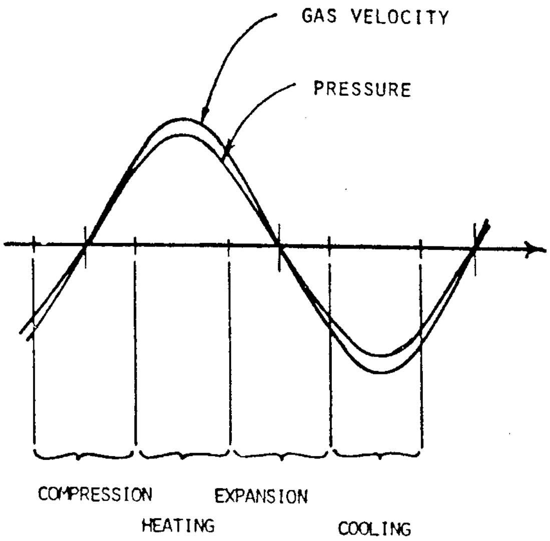


FIG. 3. Pressure and velocity as a function of time for gas through which a traveling wave is propagating. For gas in a properly orientated, differentially heated regenerator, positive velocity results in gas moving to the hot end and being heated, while negative velocity results in gas being cooled. As shown, the gas undergoes a cycle of compression, heating, expansion, and cooling, similar to that occurring in a regular Stirling engine.

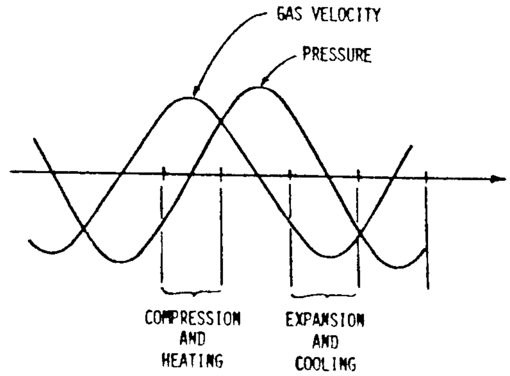


FIG. 4. Pressure and velocity as a function of time in a gas in which a standing wave exists. For gas in a differentially heated regenerator, positive velocity results in gas moving to the hot end and being heated, while negative velocity results in gas being cooled. As shown, in this cycle compression and heating occur simultaneously and expansion and cooling are also simultaneous. In order for this system to convert thermal energy into acoustical energy, a regenerator with poor heat transfer must be used to delay the heating and cooling processes so that heating follows compression and cooling follows expansion. This renders the process irreversible and fairly inefficient.

In the case of acoustical waves traveling in the reverse direction, from hot to cold through the regenerator, the flow of gas relative to the pressure wave is reversed. In this case, acoustical energy is converted into thermal energy, increasing the thermal gradient. This action allows this device to pump heat.

In a traveling wave heat pump, the waves pump heat in a direction opposite to their propagation direction: waves go from hot to cold and heat is pumped from cold to hot. In the traveling wave heat engine, the heat flow direction is still opposite to the direction of wave propagation: waves go from cold to hot and heat goes from hot to cold. This action is analogous to that occurring in a positive displacement water pump: the pump's rotational speed and direction determine the rate and direction of the water flow through it. In a traveling wave heat engine, the traveling waves determine the direction and rate of heat flow through the device. In the case of the pump, the pressure of the water on the input as compared with that on the output determines whether the pump "pumps" the water or the water "drives" the pump. Similarly, in the case of the traveling wave heat engine, the temperature of the acoustical input end as compared with the output end, determines the direction of energy transformation between acoustical and thermal forms. If the acoustical input is colder than the output (as in the case of the heat engine), the "thermal potential" drives or amplifies the waves. On the other hand, if the input is hotter than the output (as in the case of the heat pump), the "reverse thermal potential" "bucks" or attenuates the waves and draws energy from them, as is necessary for pumping heat.

The traveling wave heat engine might be considered similar to some Stirling engines which use a column of liquid as the piston.[6] The traveling wave heat engine's use of an air column as a piston should be an improvement since it lacks the problems typically associated with liquids such as confinement, maintenance of the liquid, and corrosion, and should have less viscous losses.

II. THERMODYNAMIC CYCLE OF SINGING PIPES

The pressure and velocity peaks of *standing* waves such as those in a singing pipe are phased to occur one after the other such that the waves force the gas in the temperature gradient regions of the pipes to undergo heating simultaneously with compression and cooling simultaneously with expansion (shown in Fig. 4). Such a cycle would not convert heat into acoustical power were it not for thermal delay in the heating and cooling processes which allow some of the heating to occur after the compression and some of the cooling to occur after the expansion. In fact, in a standing wave heat engine, best operation occurs when the thermal delay causes 90° phase lags[2] in the heating and cooling processes. These delays are paramount to regenerator ineffectiveness and decrease the efficiency of the device, as well as rendering it irreversible. Since the *traveling* wave heat engine does not rely on such thermal delay for proper phasing, it can capitalize on high regenerator effectiveness, is capable of higher efficiences, and is also reversible. On the other hand, the addition of a small component of properly phased standing waves to a traveling wave heat engine might improve that engine's efficiency by correcting for small inevitable thermal delays in its heat exchange processes.

III. THEORETICAL GAIN OF A TRAVELING WAVE HEAT ENGINE

In the limit of a very short regenerator ($l \ll \lambda$), a traveling wave heat engine can be modeled by a lumped parameter circuit shown in Fig. 5. The parameters I_i, I_o, and I_g are the input, output, and gain volumetric flows of the gas, and P_i and P_o are the input and output

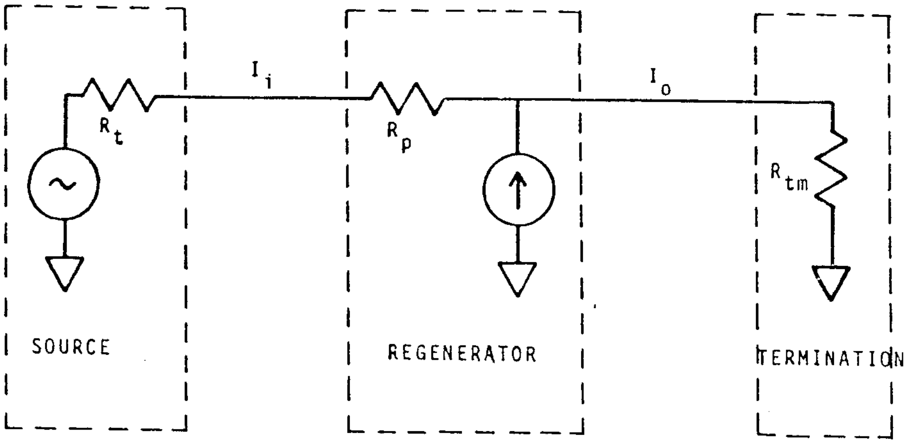


FIG. 5. A lumped parameter model of a standing wave heat engine including a source and termination of the waves.

pressures. Using the ideal gas law (so volumetric flow varies as T/P) we get the current gain of the regenerator:

$$g \equiv I_o/I_i = T_o P_i / T_i P_o = (T_o/T_i)[1-(R_p/R_t)]^{-1},$$

where T_i and T_o are the input and outlet (absolute) temperatures, the regenerator packing resistance $R_p = (P_i - P_o)/I_i$, and the impedance of the tubing $R_t = P_i/I_i$. If the packing is chosen such that $R_p \ll R_t$, we get

$$g \simeq I_o/I_i \text{ and } P_i \simeq P_o .$$

The power gain G is $G \equiv (P_o I_o / P_i I_i) \simeq g \simeq (T_o/T_i)$, and so is equal to the absolute temperature rise that the wave encounters in going from cold to hot through the regenerator.

A longer regenerator can be considered as a series of short ones:

$$G = G_1 G_2 G_3 \ldots G_N = \left(\frac{T_1}{T_o}\right)\left(\frac{T_2}{T_1}\right)\ldots\left(\frac{T_N}{T_{N-1}}\right) = \frac{T_N}{T_o} = \frac{T_o}{T_i} ,$$

as before.

Note that in practice, this gain may be reduced because of inadequate heat exchange, end effects of the regenerator, and viscous losses. Also, the gain can be reduced by impedance mismatches caused by the column and the temperature gradient. Mismatches can be eliminated by correct choice of tubing diameters and introduction of compensating reflections.

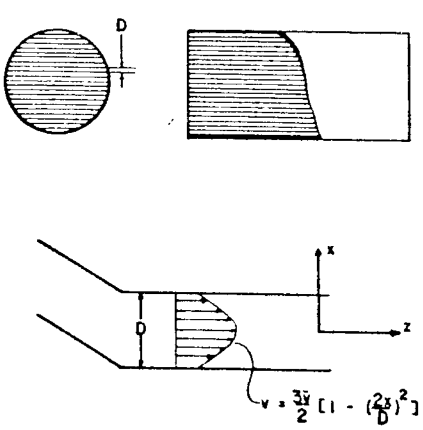


FIG. 6. Regenerator and coordinates used in calculations of power limitations. The regenerator consists of a set of parallel plates with a uniform spacing D.

IV. THEORETICAL LIMITS TO POWER DENSITY

As stated earlier, the gas flow in this engine is similar to that in a conventional piston Stirling engine. The limitation to power should also be similar to such a conventional engine. In this section we will calculate the limitation to power due to turbulence and that due to the requirement for good heat exchange between regenerator and gas. In both calculations we will assume a regenerator made of parallel plates or foils (Fig. 6) which is easily calculated analytically and is recognized as the best regenerator packing for conventional Stirling engines.[7]

A. Limitation due to turbulence

At low-power levels, the gas flow through the regenerator will be laminar and the flow resistance approximately constant. The feasibility of a machine with a reasonable flow resistance in this region will be calculated in the next section. However, near a Reynold's number ($R = \rho v D/\mu$) of 4000, the flow will become turbulent and the flow resistance will increase proportionally to the gas velocity.[8] For a regenerator spacing $D = 0.3$ mm and at STP in air, turbulent flow will occur at a sound intensity of

$$I = (\rho c)(\mu R/\rho D)^2$$
$$= (406)[(1.9\times10^{-5})(4000)/(1.29)(3\times10^{-4})]^2$$
$$= 1.56\times10^7 \text{ W/m}^2 .$$

Clearly this is not much of a limitation.

B. Limitation due to heat exchange

In this calculation we will neglect edge and regenerator end effects and assume a well developed laminar

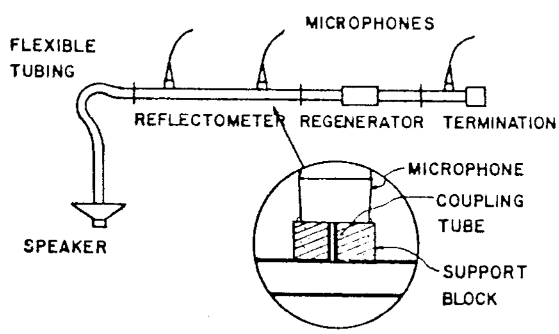


FIG. 7. Experimental setup used to measure the low-power acoustical gain of a differentially heated regenerator.

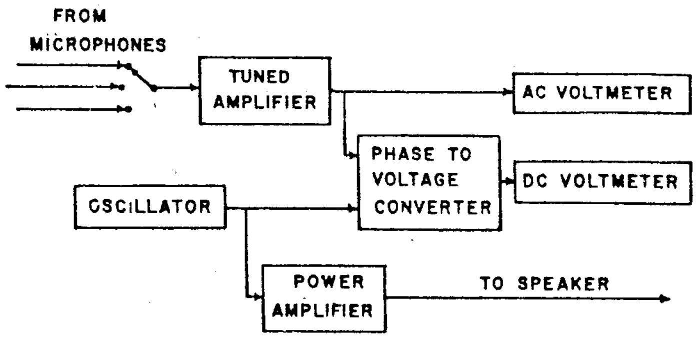


FIG. 8. Electronics used in gain measurements.

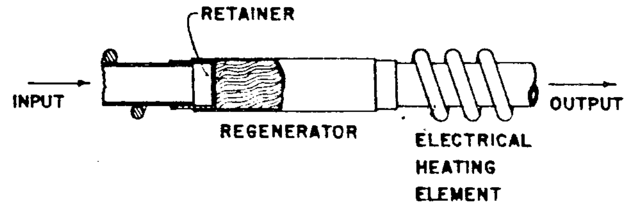


FIG. 9. Regenerator used in gain measurements. A porous retainer keeps the steel wool packing in place but is transparent to waves. The heating elements at both ends allows either end to be heated while the other remains cool. Silicon diode temperature probes are attached to the regenerator ends to monitor the temperatures there.

flow with a well developed temperature profile. The following calculation will consider the flow in one channel of the regenerator of Fig. 6. The velocity is given by

$$v = \tfrac{3}{2}\bar{v}[1 - (2x/D)^2]\,, \tag{1}$$

where $\bar{v}$ is the average velocity, x is the distance from the channel center, and D is the channel width. Neglecting longitudinal heat conduction, the temperature distribution must satisfy

$$k\frac{d^2T}{dx^2} - \rho C v\frac{dT}{dz} = 0\,,$$

where k is the thermal conductivity, ρ is the gas density, and C is the heat capacity at constant pressure. Assuming a solution of the form

$$T(x,z) = X(x) + \left(\frac{dT}{dz}\right)_w z\,,$$

where X is the deviation of the gas temperature from the wall temperature at a certain position z along the channel and $(dT/dz)_w$ is the temperature gradient along the regenerator wall, we get

$$X = \frac{3\rho C D^2\bar{v}}{8k}\left(\frac{dT}{dz}\right)_w\left[-\frac{5}{12} + 2\left(\frac{x}{D}\right)^2 - \frac{4}{3}\left(\frac{x}{D}\right)^4\right].$$

The average value of X is

$$\bar{X} = \frac{\int_0^{D/2} X\,dx}{\frac{1}{2}D} = \frac{\rho C D^2\bar{v}}{10k}\left(\frac{dT}{dz}\right)_w. \tag{2}$$

The resistance to flow is

$$R_p \equiv \frac{\Delta P}{\bar{v}A} = \frac{l}{\bar{v}A}\frac{dP}{dz} = \frac{l\mu}{\bar{v}A}\frac{d^2v}{dx^2}\,,$$

where A is the regenerator cross sectional area and l is its length. Since $d^2v/dx^2 = 12\bar{v}/D^2$ from Eq. (1), we get

$$R_p = 12\mu l/D^2A\,. \tag{3}$$

Finally, using Eqs. (2) and (3) we get that the number of heat transfer units[6] NTU is given by

$$\mathrm{NTU} = \frac{\Delta T}{\bar{X}} = \frac{5}{6P_r}\frac{R_p}{R_t}\frac{c}{\bar{v}}\,,$$

where $P_r = \mu C_p/K \simeq 0.7$ is the Prandalt number, $R_t = \rho c/A$ is the characteristic impedance of the regenerator, c is the velocity of sound, and $\Delta T = (dT/dz)_w l$ is the total temperature drop from one end of the regenerator to the other. Solving for the sound intensity and assuming a minimum NTU and R_t/R_p both equal to 5, we get

$$I = \rho c\bar{v}^2 = (\rho c)\left(\frac{5}{6}\frac{1}{P_r}\frac{R_p}{R_t}\frac{c}{\mathrm{NTU}}\right)^2$$
$$= (406)\left(\frac{5}{6}\frac{1}{(0.7)}\frac{1}{5}\frac{(330)}{(5)}\right)^2$$
$$= 100\ \mathrm{kW/m^2}\,.$$

While this is less than the limitation due to turbulence, it still makes for an attractive engine. It would mean that an engine with a 10-in. i.d. regenerator (cross-sectional area of 0.05 m^2) would be able to handle a sound power of 5 kW. With a temperature drop of 200 °F and a gain of 1.2, the engine could generate 1 kW which would be adequate for a stationary engine. If desired, higher power might be achieved with higher temperature drops, multiple regenerators, special piping and resonant transformers to increase the effective impedance of the engine, higher pressure gas, or use of helium or hydrogen as working fluids.

V. GAIN MEASUREMENTS

The acoustical gain in a differentially heated regenerator has been measured at relatively low acoustical power levels. The experimental apparatus is shown in

TABLE I. Measured low power gains for acoustical waves traveling through the regenerator shown in Fig. 9 with various temperature gradients.

Temperatures			Gain = power out/(power in−power reflected)		
Input (°C)	Output (°C)	Difference (°C)	Theoretically expected gain	Measured gain	Measured gain normalized by first entry
90	90	0	1.00	0.81	1.00
150	90	−60	0.86	0.70	0.86
90	150	+60	1.16	0.90	1.11

Fig. 7. It consists of a loud speaker with damping, a flexible tube (to carry the sound, but not wall vibrations), and a series of rigid tubes. The first tube (the reflectometer) has two identical microphones mounted on it spaced a quarter wavelength apart (for the 190-Hz sound used). Each microphone is coupled to the reflectometer tube via a 2 cm length of 1-mm i.d. tubing. The sound from the speaker passes through the flexible tube, on through the reflectometer, on through the regenerator column, and finally to a termination tube which has a third microphone and an acoustical terminating resistor. The outputs of the three microphones are connected to a switch box (see Fig. 8) which allows each in turn to be connected up to electronics which measure the amplitude and phase of each microphone signal relative to that of the signal driv ng the speaker. Use of the amplitude and phase of the st two signals allows for the calculation of the ampli es and phases of the wave traveling from speaker to regenerator (the incident wave) and the wave reflected off the regenerator (the reflected wave). The third microphone allows the calculation of the wave transmitted through the regenerator (the transmitted wave). The microphones are calibrated by removing the regenerator and first putting a plug in its place so that the incident and reflected waves are equal with a known phase relationship, and secondly by connecting the termination tube directly onto the reflectometer tube so that the incident and transmitted wave are equal and the reflected wave zero. After calibration, the regenerator is inserted and its effects on the waves are measured. It is important to measure the reflected wave because the regenerator and temperature gradient in it can cause reflections which can affect the gain of the regenerator.

The regenerator used is shown in Fig. 9. It was packed with 00-grade steel wool. Heating coils were used to apply heat to either end of the regenerator. Temperatures of the regenerator ends were measured by attaching small calibrated silicon diodes to the ends and measuring the diodes' forward voltage drop.[9]

Table I shows the results. Listed are the theoretically expected gain, the measured gain [power out/(power incident–power reflected)], and the relative measured gain, all three with no differential heating, with heating of the acoustical input end, and finally, with heating of the acoustical output.

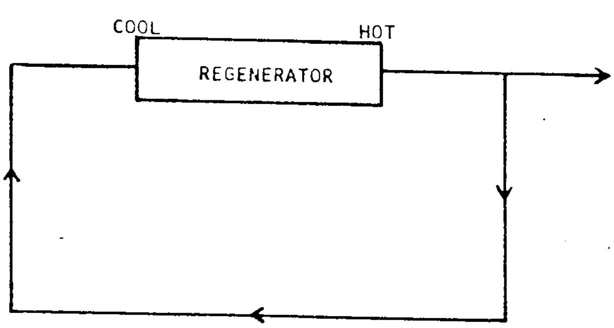


FIG. 10. An acoustical oscillator consisting of a traveling wave heat engine and a positive feedback pipe. Some sound can be coupled out to power an electrical transducer and make electricity, for example.

With no temperature gradient, the gain was 0.81, indicating a 19% attenuation of insertion loss due to flow resistance. A –60 °C temperature gradient (sound going from hot to cold), and a +60 °C gradient produced gains of 0.70 and 0.90, respectively. When compared with the 0.81 no gradient transmission, they had relative gains of 0.81 and 1.11, which are very close to the theoretically expected gains of 0.86 and 1.16. These results indicate that the heat transfer in the regenerator was more than adequate for the power levels used (probably about 80 dB), and that less dense packing should be used which will have smaller insertion loss.

With a better choice of packing, and perhaps a greater temperature gradient, we expect to be able to achieve absolute gains of greater than 1.0. With such gains, we could make an acoustical oscillator and experiment at very high acoustical power levels, such as those used in practical engines. Achieving this, we hope to measure the efficiency and power density actually achievable in these traveling wave heat engines.

VI. APPLICATIONS OF TRAVELING WAVE HEAT ENGINES

A traveling wave heat engine is an acoustical amplifier, which can be used in various acoustical circuits, for a range of energy conversion applications. For example, Fig. 10 shows an acoustical oscillator consisting of such an engine and a pipe to provide positive acoustical feedback. Such a device would convert thermal energy into acoustical power which could be further converted into electrical power via an acoustical-to-electrical transducer, similar to the earlier Bell Labs singing pipe device, but with higher efficiency because of its use of traveling waves. The lack of moving parts, simplicity, and reliability of this device may make it attractive as a power supply for isolated equipment. Figure 11 shows another circuit, a thermally driven heat pump, which consists of a pair of traveling wave heat engines, one converting heat into acoustical power which is fed through the second engine acting as a heat pump. This device might find use as a solar-

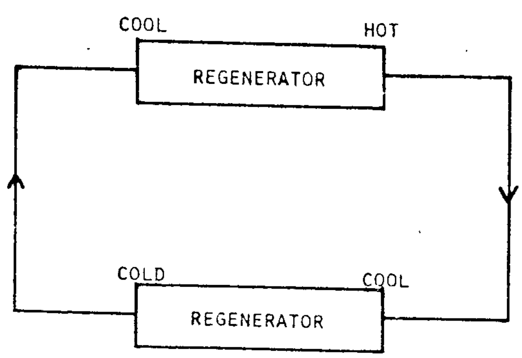


FIG. 11. A thermally driven heat pump consisting of two standing wave heat engines. An external heat source provides thermal energy to drive the top engine, which produces the acoustical waves to power the bottom engine acting as a heat pump. The combination might be used as a refrigerator, air conditioner, or heat pump. The lines represent pipes with the arrows showing the direction of wave propagation.

driven air conditioner where its simplicity, low cost, and long lifetime may be more important than its low specific power. Obviously, many other circuits and applications involving conversion of thermal and mechanical power are possible.

VII. THE TRAVELING WAVE HEAT ENGINE—A GENERALIZED STIRLING ENGINE

Since the motion and pressure cycles of the gas in the regenerator of a normal Stirling engine are identical with those occurring in the regenerator of a traveling wave heat engine,[5] one might consider a normal Stirling engine to be a special case of a traveling wave heat engine with its "waves" being created and absorbed by moving pistons. Thus, one could consider the traveling wave heat engine, consisting of a differentially heated regenerator which has the appropriate pressure and motion cycles characterized by a traveling wave, to be the essence of a Stirling engine or to be a generalized Stirling engine, with the method for creating and using these waves to be specified for the particular type of Stirling engine. The use of waves to characterize the pressure and motion cycles might be particularly illuminating in Stirling engines operating at high cyclic rates where the wavelengths become short and the transit responses and delays important. The transit responses and delays can be treated as part of the wave propagation process and can be dealt with using the mathematical methods and experimental techniques developed for acoustics and microwaves.

[1]R. V. L. Hartlay, U. S. Patent No. 2,549,464 (1951).
[2]W. A. Marrison, U. S. Patent No. 2,836,033 (1958).
[3]P. H. Ceperley, U. S. Patent No. 4,114,380 (1978).
[4]*Should We Have a New Engine?*, SPL SP 43–17, JPL (California Institute of Technology, Pasadena, CA, 1975).
[5]G. Walker, Sci. Am. **234**(8), 80 (1973).
[6]W. R. Martini, "Developments in Stirling Engines", presented at the Annual Winter Meeting of ASME, MDAC Paper WD 1833, Nov. 1972 (unpublished).
[7]W. R. Martini, "The Free-Displacer, Free-Piston Stirling Engine. . . Potential Energy Conserver," IECEC 1975 Record, 995 (1975).
[8]*Noise and Vibration Control*, edited by L. L. Beranek (McGraw-Hill, New York, 1971), p. 397.
[9]C. J. Koch, Electronics **51**, 110 (1976).

VOLUME 50, NUMBER 7 PHYSICAL REVIEW LETTERS 14 FEBRUARY 1983

p. 499-502

Experiments with an Intrinsically Irreversible Acoustic Heat Engine

John Wheatley, T. Hofler,[a] G. W. Swift, and A. Migliori
Los Alamos National Laboratory, University of California, Los Alamos, New Mexico 87545

(Received 8 November 1982)

The general qualities of a type of thermodynamic engine that depends intrinsically for its operation on irreversible processes are set forth and demonstrated experimentally in the context of a thermoacoustic heat-pumping engine.

PACS numbers: 44.60.+k, 43.25.+y

When a stack of plates as in Fig. 1 is placed near the end of a closed tube containing ^{4}He gas and an oscillating pressure is applied to the gas with period comparable to the thermal relaxation time between gas and plates, starting with a uniform temperature throughout, the end C of the stack cools while the end H heats. This is a consequence of an acoustically stimulated average hydrodynamic flow of entropy in the gas from the end C to the end H of the stack. If the stack is weakly thermally coupled to its surroundings at C and H and otherwise thermally insulated, a limiting temperature distribution along the stack quickly results, roughly independent of the static and dynamic pressure of the gas and of the frequency. This temperature distribution can be stabilized by manipulating the thermal contacts at the ends. The temperature difference thus obtained is geometry dependent but independent of other engine parameters and can be very large compared with the adiabatic gas temperature oscillations resulting from the pressure oscillations. For a given temperature T_H at H, the limiting temperature distribution for zero heat transferred to a thermal reservoir at H bounds a region of generally lower temperature gradient in temperature-position space where work is used to effect the transfer of heat. There is another limiting temperature distribution, for the same T_H, corresponding to a rather larger

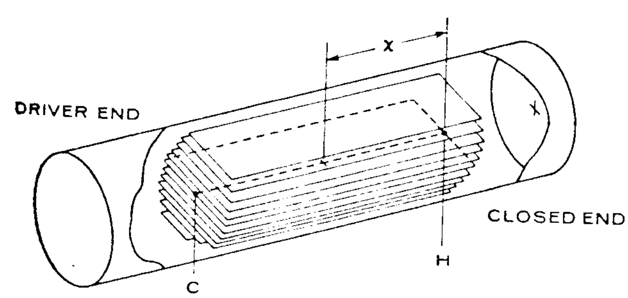


FIG. 1. Schematic of an intrinsically irreversible thermoacoustic engine.

temperature difference imposed on the stack which bounds a region of generally higher temperature gradient where heat is absorbed at H and work may be produced in the form of spontane-

ous oscillations. Related phenomena were investigated as early as the last century by Sondhauss[1] and discussed qualitatively by Lord Rayleigh[2] in the case of the spontaneous oscillations where heat is converted to work, and more recently by Gifford and Longsworth[3] with their "pulse-tube refrigerator," in the case where work is used to pump heat. Further background has been given by Rott,[4] whose thermoacoustic theory facilitates a quantitative interpretation of these phenomena, in particular the recent quantitative work on the spontaneous oscillations in ^{4}He gas by Yazaki, Tominaga, and Narahara.[5] In this Letter we present some of the central experimental characteristics and calculated properties of our acoustic heat-pumping engine. Although this engine is thermoacoustical in nature, we believe that it is just one manifestation of a quite general class of intrinsically irreversible thermodynamic engines which need bear no relation to acoustics and which can use working substances other than gases. We also propose some of the general features of such intrinsically irreversible engines.

Both the entropy flow central to the operation of this engine and the relationship to acoustics are demonstrated very nicely by a simple instrument which we call a thermoacoustic couple, shown in the inset to Fig. 2. It is just a short "engine" of length Δx very much less than the radian length λ of the sound and consisting of one or more solid plates. The central plate of the couple is fitted with a thermopile to measure $\Delta T = T_C - T_H$. This temperature difference can be used to measure entropy flow in the gas provided the longitudinal thermal conductance Σ of the couple is so high that it dominates all diffusive heat flows. In the presence of acoustic power, entropy flows out of one end of the plates, down along the plates in the gas, and back into the other end. The corresponding second-order energy flow is, following Rott,[4]

$$\langle\Delta \dot{H}\rangle = \Pi \int dy\, \rho_m T_m \langle s_1 u_1\rangle = \Pi \int dy\, \langle(\rho_m c_p T_1 - p_1) u_1\rangle, \quad (1)$$

where Π is the surface area per unit length, or perimeter; y is the direction perpendicular to the plates; ρ_m and T_m are mean mass density and mean temperature; s_1, u_1, T_1, and p_1 are, respectively, the first-order entropy per unit mass, velocity, temperature, and pressure; c_p is the constant-pressure specific heat; and the angular brackets indicate a time average. In the boundary-layer approximation $(\rho_m c_p T_1 - p_1)$ is zero except in the thermal boundary layer of characteristic dimension $\delta_\kappa = (2\kappa/\omega)^{1/2}$ next to the plates, where κ is thermal diffusivity and ω is angular frequency. In the absence of the couple, $\langle\Delta \dot{H}\rangle$ would be zero in the space occupied by it. Evaluation

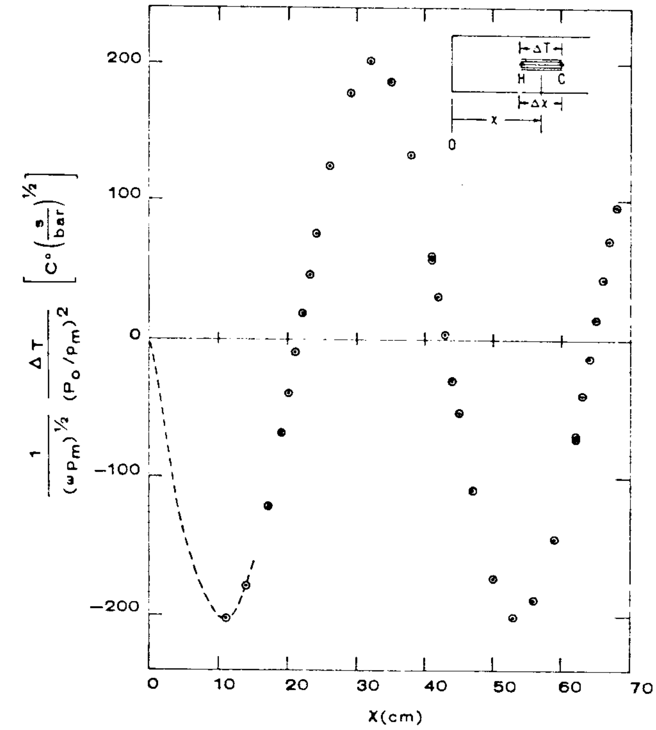


FIG. 2. Normalized thermoacoustic couple response $(p_m\omega)^{-1/2}\Delta T/(P_0/p_m)^2$ as a function of the average distance x to the couple from the closed end of a tube filled with ^{4}He gas at 2.55 bars and 22 °C and resonant at 1163 Hz. The normalization of ΔT gives a quantity which is independent of static and dynamic pressure and frequency and depends only on the geometry and constitution of the couple. The schematic geometry is shown in the inset: This couple had five parallel plates 2.0 cm long and 1.9 cm wide with a six-couple thermoelectric pile measuring ΔT across the central plate. Each plate was a composite of two 0.0125-cm-thick stainless-steel sheets with a 0.0125-cm-thick fiberglass sheet sandwiched between them. The plate separation is 0.114 cm while $\delta_\kappa = 0.014$ cm. The data were obtained with $P_0/p_m = 0.0025$.

of (1) for a large enough Σ that temperature gradients are adequately small gives, in the boundary-layer approximation and with the assumption that the phase between p_1 and u_1 is $\pi/2$ far from the plates,

$$\langle\Delta \dot{H}\rangle = -\tfrac{1}{4}\Pi p_a v_a \delta_\kappa\left(\frac{1+\sqrt{\sigma}}{1+\sigma}\right) = -\frac{\Pi}{4}\frac{\omega\delta_\kappa P_0^2}{\gamma p_m}\left(\frac{1+\sqrt{\sigma}}{1+\sigma}\right)\left(\frac{a}{2\omega}\right)\sin\left(\frac{2\omega}{a}x\right). \quad (2)$$

Here p_a and v_a are the pressure and velocity amplitudes far away from the plates, γ is the ratio c_p/c_v of specific heats, σ is the Prandtl number, p_m is the mean pressure, P_0 is the amplitude of the dynamic pressure measured at the closed end of the tube, a is the sound velocity, and x is the mean distance of the couple from the closed end of the tube. We assume that the response ΔT of the couple may be determined from

$$-\Sigma\Delta T + \langle\Delta \dot{H}\rangle = 0, \quad (3)$$

where $\langle\Delta \dot{H}\rangle$ is evaluated in the center of the couple. This assumes that the average energy

flow down the couple in the gas in the thermal boundary layer is balanced by diffusive heat flow in the plates. An example of normalized thermoacoustic couple response is shown in Fig. 2, the couple support structure having prevented measurements near the closed end. If enough plates giving a high enough Σ are used there is quantitative agreement between the observations and Eqs. (2) and (3) for a broad range of P_0 and p_m, provided P_0/p_m is sufficiently small. The response is zero at both pressure and velocity antinodes; the hot end of the couple is always toward the nearest pressure antinode.

For a stack substantially longer than a thermoacoustic couple but still shorter than λ, the entropy flow in the gas will rapidly lead to a large temperature difference between H and C if longitudinal heat flow is small and thermal coupling at the ends is weak. We have studied this quality extensively; details will be presented elsewhere. Briefly, in the experiments the temperature at the closed end is maintained near ambient while the temperature at some distance from C toward the driver is controlled independently to be some value T_C'. Under these conditions acoustic power produces a temperature distribution in the stack which is "rigid" in the sense that, provided the longitudinal heat transfer is small, it does not depend strongly on the temperature of either the closed end or the controlled region. This temperature distribution changes by only a few percent as p_m and P_0^2 are changed by an order of magnitude and ω by a factor of 3. The controlled temperature T_C' can be adjusted to make T_H approximately ambient. For that condition we find that the normalized temperature distribution T/T_H along the stack depends primarily on the geometry, specifically on the ratio of the distance along the stack to the effective length of the open space between H and the closed end. It depends only weakly on changing the composition of the stack from longitudinal fiberglass parallel plates to longitudinally insulated *transverse* copper screens. Using a computational method based on the theory of Rott,[6,7] we find that the actual temperature distribution lies between those calculated for the two limiting cases $\langle \dot{Q}_H \rangle = 0$ and $\langle \dot{Q}_C \rangle = 0$, where these are the average heat transfer rates at H and C, respectively. The smallest value of T_C/T_H thus far observed for these conditions is about 0.53, corresponding to relative acoustic pressure amplitudes P_0/p_m of order 0.02 to 0.04. Such a large temperature difference suggests that useful devices might someday be built based on these principles.

We have not yet made measurements of the efficiency η of the acoustic engine. To develop intuition on this problem it is useful to consider the inviscid case $\sigma = 0$. For such an ideal intrinsically irreversible engine using an ideal gas, we find the result that η depends primarily on geometry and on γ rather than on absolute temperature. For an ideal irreversible magnetic engine of the present type and employing Curie-law materials, we find that η depends only on the configuration of magnetic materials in magnetic fields and the field themselves. If these ideal engines are operated near the limiting temperature distribution so that even at the nonzero frequency of the engine the heat-transfer rates approach zero, η can be shown by calculation to approach the Carnot value as an upper bound.

A few general features of this type of engine can be learned from the thermoacoustic engine. Referring to Fig. 1 our engine has a primary thermodynamic medium, ^{4}He gas, enclosed in a tube also containing a secondary thermodynamic medium, a stack of thin fiberglass plates, whose function is to exchange heat with the gas. The secondary medium should have a low longitudinal conductance to reduce longitudinal heat flows. The primary medium has a reciprocating motion with respect to the secondary medium; attendant on that motion is a thermodynamic effect, in this case a change of temperature of the gas caused by the pressure change. The processes involved in the contact between primary and secondary media must be irreversible; in the gas engine the most important process is the exchange of heat. This intrinsic irreversibility in the engine has the purpose of providing for a suitable phasing between motion and the thermodynamic effect. In the present engine there is no useful average energy flow either in the isothermal case, in which the spacing between plates is small compared with the thermal penetration depth, or in the adiabatic case, in which the spacing is large compared with the penetration depth. Finally, what is necessary to produce a cooling effect starting from zero longitudinal temperature gradient is that in some region there be an increase of energy flow in the direction of the energy flow. In the apparatus of Fig. 1 this is achieved where the plates begin on the driver side as a consequence of a rapid change there of the instantaneous heat transfer per unit length between the media. There is a corresponding heating effect at the other end of the plates. These effects are a consequence of what we call a "broken thermodynamic symmetry" between the media. If the energy flow $\langle \dot{H} \rangle$ is constant in space then we have thermodynamic symmetry. Referring for example to Eq. (2), which gives the entropy-flow part of $\langle \dot{H} \rangle$, the thermodynamic symmetry can be broken if any of the geometric quantities like Π, or thermal quantities like δ_κ, or dynamic quantities like p_a or v_a are spatially dependent. For engines in which gas is the primary medium, such as the present engine, the pulse tube of Gifford and Longsworth,[3,4] and the uniform-diameter resonance tube of Merkli and Thomann,[8]

the above conditions are met; thermodynamic symmetry, however, is broken quite differently in each case. The traveling-wave acoustic Stirling engine of Ceperley[9] is intrinsically reversible and hence of a different type.

The possibilities for the thermodynamic media and configurations for the general type of thermodynamic engine described above are very broad. Essentially any materials for which an adequate thermodynamic effect can be achieved can in principle be used. It is not necessary for the media to be separated in space; interpenetrating media, as for example the electron and ion systems in a plasma, would be appropriate. The frequency of optimum operation is characterized by the reciprocal of the natural interaction times of the media and hence might vary from, say Hz for a magnetic engine to a very high frequency for a plasma engine.

Real irreversible heat engines have also been considered recently.[10-12] In this work the Carnot engine is central, and calculation of properties of real engines are made by explicitly introducing irreversibilities external to the Carnot engine. We would like to suggest that the ideal irreversible engine of the present type, in which the irreversibilities are an inseparable part of the engine's operation, and in which the thermodynamic cycle is determined "naturally" at a finite period, would be a suitable vehicle for theoretical investigation. We conjecture that configuration and certain physical properties of the media may be primarily important while the temperature spanned follows from the power output.

We are indebted to S. L. Garrett for helping us in many ways to begin the acoustic work and to L. J. Campbell for his help in the thermoacoustic analysis.

[a]Also at: Department of Physics, University of California at San Diego, La Jolla, Cal. 92093.

[1]C. Sondhauss, Ann. Phys. Chem. 79, 1 (1850).

[2]Lord Rayleigh, *Theory of Sound* (Dover, New York, 1945), Sec. 322i.

[3]W. E. Gifford and R. C. Longsworth, *International Advances in Cryogenic Engineering* (Plenum, New York, 1966), Vol. 11, p. 171.

[4]N. Rott, Adv. Appl. Mech. 20, 135 (1980), and references therein.

[5]T. Yazaki, A. Tominaga, and Y. Narahara, J. Low Temp. Phys. 41, 45 (1980).

[6]N. Rott, Zh. Angew. Math. Phys. 20, 230 (1969).

[7]N. Rott, Zh. Angew. Math. Phys. 26, 43 (1975).

[8]P. Merkli and H. Thomann, J. Fluid Mech. 70, 161 (1975).

[9]P. H. Ceperley, J. Acoust. Soc. Am. 66, 1508 (1979).

[10]B. Andresen, P. Salamon, and R. S. Berry, J. Chem. Phys. 66, 1571 (1977).

[11]D. Gutkowicz-Krusin, I. Procaccia, and J. Ross, J. Chem. Phys. 69, 3898 (1978).

[12]P. Salamon, A. Nitzan, B. Andresen, and R. S. Berry, Phys. Rev. A 21, 2115 (1980).

Physics Today, January 1984

Intrinsically Irreversible Acoustic Heat Engine

Fill a tube, having one closed end, with some gas; close it with a piston and make the piston vibrate. What happens is exactly what you would expect—the walls of the tube heat up. But when you put a stack of thermally insulating plates, spaced apart by, say, a millimeter, near the closed end of the tube and cause the piston to vibrate at acoustic frequencies, what happens is surprising. The end of the stack nearest the piston (where the gas motion and viscous heating happen to be greatest) starts to cool, while the opposite end of the stack starts to heat, and if the stack is thermally isolated, the temperature difference across the stack between piston-end and closed-end can in a few minutes get quite large, perhaps 100°C. Producing a thermal effect such as this while having to move only one part is both remarkable and potentially useful.

The significance of the work from a scientific standpoint, however, is that the above acoustic device provides, in one case, the general principles of a new class of cyclic heat engines [illegible]hich can use any suitable thermally active materials in which [illegible] thermal cycle is determined naturally through qualities like g[illegible]metrical configuration and imperfect thermal contact. In the field of heat, especially of heat engines, it is presumptuous to imagine finding something new. Perhaps it is best to say that the acoustic experiments have led to recognition of a principle of quite general validity.

Scientists at the Los Alamos National Laboratory have recently made progress toward a working model[1] of an "intrinsically irreversible acoustic heat engine." Although the words "heat engine" usually evoke images of the steam engine, in the present context they describe an apparatus in which the concepts of heat, work, energy, and temperature are important. Some heat engines are used to turn heat into work, as in the classical steam engine; others are used to move heat from cold to hot by expending work, as in the refrigerator or the heat pump. The Los Alamos experiments thus far employed the acoustic heat pumping mode, although the same apparatus can be used, in principle, in a heat-to-cool mode by applying a large enough temperature difference.

For the words "intrinsically irreversible" we could substitute the word "natural." "Reversible" and "irreversible" are words from thermodynamics that are used to describe whether or not a given process, such as motion or heat transfer, can be reversed by an infinitesimal change in the drive. The ideal engine of Sadi Carnot (1824) is reversible and is the most efficient heat engine possible operating between two temperatures. But no real engine operating with a finite cycle time can be truly reversible. So all real engines are irreversible. What distinguishes the Los Alamos engine is that it must operate with a finite period; it ceases to function altogether as the processes become reversible. That is why it is called intrinsically irreversible.

In the acoustic engine, the irreversible process of thermal conduction, or thermal lag, introduces the concept of phase, or of "timing," into the engine. Timing in a gasoline engine means producing a spark when the piston is near top dead center. Thermal lag performs the same conceptual function in the acoustic engine except that only the piston's position is externally controlled.

In the acoustic engine, exchange of heat between the gas and the stack of plates (or really, of any other suitably porous medium) is indispensable to the engine's operation. This is a quality shared with Robert Stirling's (1816) intrinsically reversible engine. Finally, what is essential to the primary cooling and heating is the rapid change of thermal contact between gas and plates as the gas oscillates in and out of the ends of the plates; we say that this is an example of "broken thermodynamic symmetry."

So, if you cause the reciprocating mutual motion of two thermally active mediums in imperfect contact to produce thermal changes, and if you take care to break the thermodynamic symmetry, you will produce an interesting and possibly useful thermal effect in your intrinsically irreversible heat engine.

John Wheatley, Los Alamos Scientific Laboratory

1. John Wheatley, T. Hofler, G. W. Swift, and A. Migliori, Phys. Rev. Lett. 50, 499 (1983).

Natural engines

'Taconis oscillations,' the oscillations of some variable stars, and a novel form of engine are all based on cycles that involve an intrinsically irreversible process and a broken thermodynamic symmetry.

John Wheatley and Arthur Cox

Under certain circumstances, a flow of heat through a system can give rise to acoustic oscillations, converting some of the heat to work. Natural vibrators maintained by heat flows have been studied since the 1770s. Some of the best-known examples come from acoustics: the "singing flames" first investigated by Byron Higgins in 1777, the Sondhauss[1] tube and the Rijke tube.[2] Most experimenters in cryogenics have observed the "Taconis oscillations" that occur when a tube, closed at the top, is inserted into a liquid-helium dewar. A group at Tsukuba has studied[3] such oscillations quantitatively. Oscillations driven by heat also occur on a very large scale, in certain classes of variable stars.

One may also conjecture that thermodynamic engine effects occur on a scale intermediate between the atomic and the macroscopic. To develop this conjecture, the Los Alamos thermoacoustics group has studied[3] the properties of the "engine" shown in figure 1. This engine is intended to model a chain of suitably large coupled molecular vibrators. It actually consists of a chain of coupled nonlinear acoustic vibrators, filled with argon gas under pressure and driven from a common source at about 300 Hz, somewhat below the low-amplitude resonant frequency. At a sufficiently large driving amplitude, stationary localized, or solitary, states of relatively higher vibrational intensity appear on the chain. Heat is pumped toward the localized states along the surfaces of the tubes that couple the vibrators.

The basic mechanism of these engines was explained[5] qualitatively by Lord Rayleigh in his book on the theory of sound: "In almost all cases where heat is communicated to a body, expansion ensues, and this expansion may be made to do mechanical work. If the phases of the forces thus operative be favorable, a vibration may be maintained."

The historical examples we gave were of engines that convert heat to work. The alternative thermodynamic function, the transfer of heat from one place to another as a consequence of the performance of work, that is, heat pumping, is illustrated[6] in figure 2. Both processes are illustrated on the cover of this issue; here the work produced in one part of the engine is used to drive a refrigerator (an identical engine operating backwards), which cools the bulb at the bottom of the photo.

P. Merkli and H. Thomann demonstrated[7] acoustic cooling in open tubes, and William Gifford and Ralph Longsworth exibited[8] cooling in their "pulse tube," which uses a low-frequency articulated cycle with large pressure changes. The engine shown in figure 2 consists of a stack of plates placed near the end of a closed tube. When the tube is driven at acoustic resonance, a temperature difference develops along the plates: The stack heats up near the closed end of the tube and cools at the end facing the driver. Conversely, if one imposes a large enough temperature difference along the plates, the same structure generates sound rather than being driven by it.

These engines can be quite effective. The graph in figure 2b shows the response of the engine shown in figure 2a to a 411-Hz tone whose dynamic pressure amplitude is a bit less than 1% of the ambient pressure. The tube is filled with helium at about five atmospheres pressure. Within a few seconds a five-degree temperature difference develops along the stack of plates. For a larger pressure amplitude, the effect is even more striking. At a dynamic pressure amplitude of 4% of the ambient pressure, a 100-C° temperature difference develops along the plates in one minute.

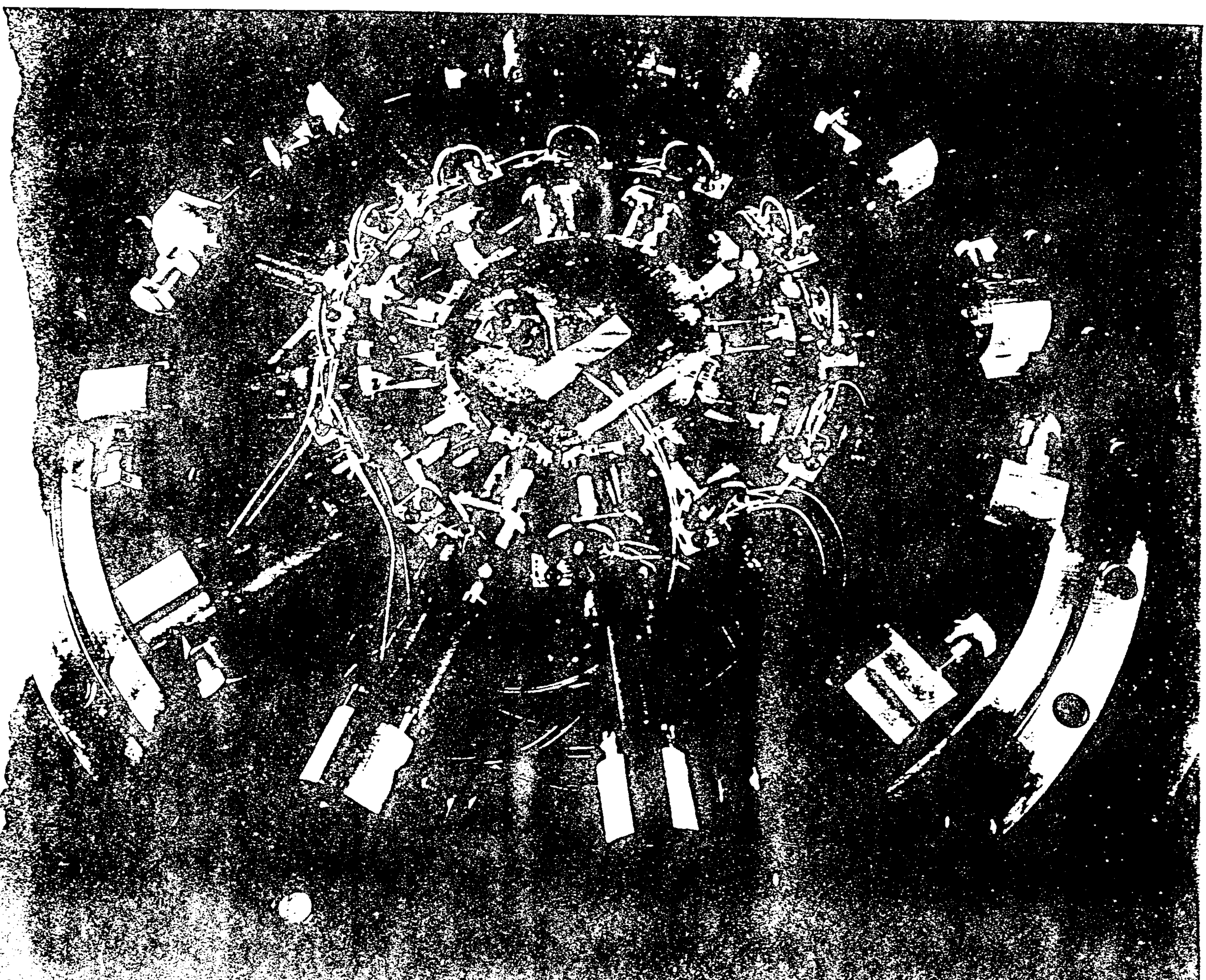
The graph of figure 2b also shows that the primary effect occurs at the ends of the plates. Thermocouples placed into the center of the stack show hardly any variation of temperature. As we shall see when we discuss the details of the engine's operation, the breaking of the translational symmetry at the ends of the plates is crucial to the operation.

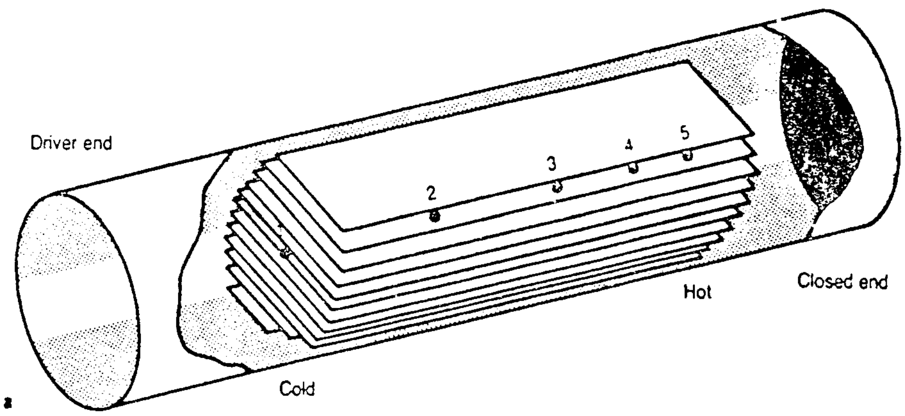
The engine shown in figure 2 has another property that distinguishes it from familiar thermodynamic cycles, such as the Carnot cycle (isothermal–adiabatic–isothermal–adiabatic). Its operation is based on the exchange of heat between the plates and the gas through a nonzero temperature difference. In fact, the engine will neither pump heat nor do work if all processes are reversible: It is intrinsically irreversible. In this, the engine is not unique; both the Otto cycle (isochoric–adiabatic–isochoric–adiabatic, an idealized gasoline-engine cycle) and the Brayton cycle (isobaric–adiabatic–isobaric–adiabatic) are also intrinsically

John Wheatley is a University of California–Los Alamos fellow at Los Alamos National Lab and professor of physics at the University of California, Los Angeles. Arthur Cox is a laboratory fellow at Los Alamos.

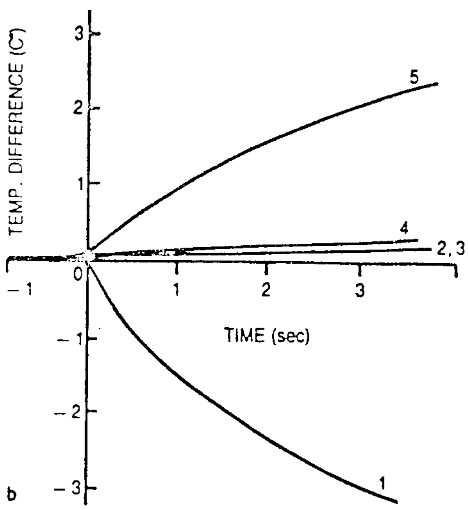
 p. 50-58 0031-9228 / 85 / 0800 50- 08 / $01.00

The 'dodecagon,' an engine intended to model vibrations in large molecules. Each of the 12 identical nonlinear acoustic vibrators consists of two bulbs coupled by a neck that narrows as the amplitude of oscillation increases (as the throat constricts in coughing); the inner bulbs are coupled to their nearest neighbors and are driven by acoustic vibrations from the center of the array. The coupling tubes contain a single plate of the sort described in figure 3. Filled with argon and driven at around 300 Hz, the dodecagon exhibits self-focusing vibrational excitations. Figure 1





Thermoacoustic engine or heat pump. The sketch, **a**, shows the arrangement of plates near the closed end of a driven organ pipe. Thermocouples are placed along the central axis of the stack as shown. The graph, **b**, shows the temperature as a function of time, after acoustic oscillations start at $t = 0$, with no initial temperature gradient. Note that the two ends cool or heat rapidly, while the central portion of the stack is not greatly affected. Figure 2



irreversible, but for quite different reasons. (Some readers will see vestiges of the Otto and Brayton cycles in our heuristic description below of the engine of figure 2 in an articulated cycle.)

The natural engine

We will use the term "natural engine" to describe in general objects that work in the manner of the device shown in figure 2. To be precise, a natural engine is an object that functions either to perform external work as a consequence of energy flows or to transfer heat from one place to another as a consequence of work being performed on it, when the following ingredients are present:

▶ There are two or more thermodynamic mediums in a reciprocating relative motion that is attended by a thermodynamic effect.

▶ Time phasing of the thermodynamic effect with respect to the relative motion is achieved by some *natural irreversible* process (such as thermal conduction across a temperature difference).

▶ The thermodynamic symmetry along the direction of relative motion is broken in some way—by geometry or configuration, as in the heat pump of figure 2; by dynamics, as in the nonlinear engine of figure 1; or by changes of material properties, as in one of the primary mechanisms responsible for stellar pulsations.

The term "natural" does not fully describe such an engine, but it has the advantage of being short and of referring to the crucial role played by natural irreversible processes in determining the phasing in the engine.

The notion of improving the performance of heat engine cycles by introducing a second thermodynamic medium, in addition to the primary medium, was introduced in 1816 by Robert Stirling. In the engine of figure 2 the primary medium is helium gas, the secondary medium is the stack of plates, and the thermodynamic effect is the adiabatic change of temperature with pressure in the gas. The timing of the various processes in a cycle is central to the operation of all engines that are to produce some net work over a cycle. In the engine of figure 1 the timing is provided by the thermal lag of the plates with respect to the gas; the engine requires no externally controlled timing mechanism, unlike the usual engines such as the internal-combustion engine.

The concept of a natural engine is often most useful when primary and secondary mediums are extended along the axis of relative reciprocating motion, so that, effectively, there are many elementary "engines" in series. The energy, work and heat flows along this direction are modified by the lateral thermodynamic exchanges between the mediums. Whenever this lateral contact changes, the time-averaged energy flow along the axis of relative motion also changes, and we say that the thermodynamic symmetry has been broken. In the device of figure 1 the thermodynamic symmetry is broken at the ends of the stack. This is where the interesting effects occur.

Our definition for a natural engine is very general; we have left unspecified the mediums, the time scale, the nature of the contact, the irreversibility and the configuration of the devices. (Of course, it represents only a subclass, though a very useful one, of all of nature's engines.) For example, one can readily imagine[6] magnetic engines working on these principles. However, some of the simplest natural engines are to be found in acoustics. Most of the quantitative understanding of thermoacoustic effects in gases is based on the theoretical work of Nikolaus Rott.[9]

In the remainder of this article we begin by discussing the physics of the elementary thermoacoustic engine, and continue with a description of three potentially practical devices. Then we show how the concepts apply to some very large vibrators maintained by heat—certain variable stars.

Elementary thermoacoustic engines

We begin by discussing the thermodynamics of a parcel of gas (first medium) moving reciprocally (compressing and expanding) with respect to a wall (second medium), as shown in figure 3. The parcel makes thermal contact with the side wall; the end wall determines the directions of relative motion that correspond to compressions and expansions. Note first that if the parcel moves reversibly, either very rapidly (adiabatically) or very slowly (isothermally), there is no net work and no net heat flow in a cycle. Any interesting effect is thus a consequence of irreversibility, and it suffices to calculate work and heat flows associated with the irreversible processes. (We will neglect the effects of viscosity.)

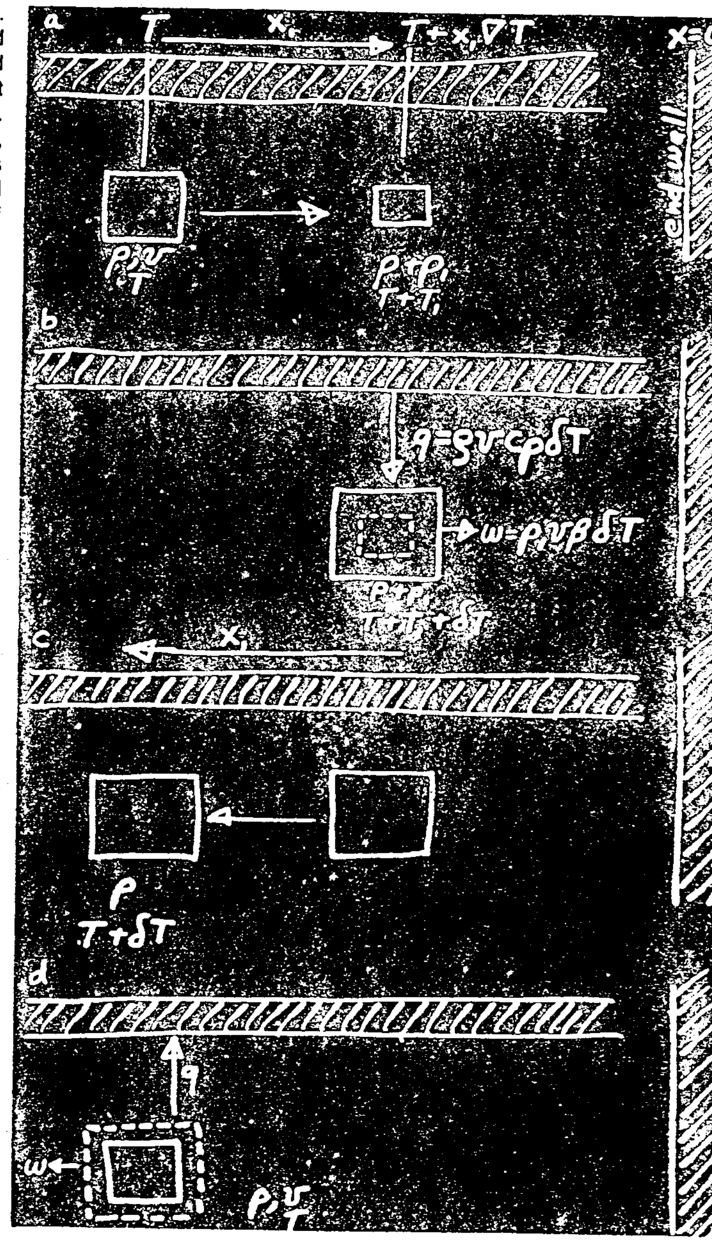
Imagine a rapid (reversible adiabatic) compression during which the parcel moves a distance x_1 toward the end wall and increases in temperature by T_1. During a suitable time lag, the parcel again makes thermal contact with the wall, which we imagine has some longitudinal temperature gradient ∇T. The excess temperature δT of wall over parcel just after compression is then

$$\delta T = x_1(\nabla T) - T_1$$

For positive δT, heat flows into the parcel and it expands (figure 2b). The fractional expansion of the parcel is $\beta\delta T$, where β is the thermal expansion coefficient. The heat that flows into the parcel is

$$q = v\rho c_p \delta T$$

Heat transport by acoustic vibrations. The diagrams show how a standing sound wave in a tube (with a velocity node and pressure antinode at $x = 0$) transports heat and energy along a plate inside the tube. During each cycle of acoustic oscillations, a parcel of gas is compressed as it is pushed a distance x_1 toward the closed end; at its new position it absorbs heat (if the temperature gradient in the plate is large enough: "superadiabatic"); in the expansion cycle of the oscillation, the gas returns to its initial position, where it is now hotter than the plate, and rejects the excess heat it absorbed earlier. Figure 3



where v is the volume of the parcel and ρc_p is the specific heat per unit volume. The work the parcel does on its surroundings is

$$w = p_1 v \beta \delta T$$

where p_1 is the increase in pressure of the gas arising from its displacement and compression.

For negative δT, heat flows out of the parcel to the wall and work is done on the parcel by its surroundings. For a cyclic process the former case corresponds to heat being transferred from a higher to a lower temperature, a distance x_1, and to external work being done; the latter case corresponds to heat being transferred from a lower to a higher temperature at the expense of external work.

These two contrasting heat engine functions are separated by the case $\delta T = 0$, where the temperature change $x_1(\nabla T)$ in the wall just matches the adiabatic temperature change T_1. The critical temperature gradient separating work-producing and work-absorbing ideal engines is then

$$(\nabla T)_{\text{crit}} = T_1/x_1$$

It is helpful in summarizing these results to define a quantity

$$\Gamma = \nabla T/(\nabla T)_{\text{crit}}$$

If the plates are near the closed end of the tube and if their total distance x from the closed end is much smaller than the radian length $\lambda/2\pi$ of the acoustic oscillations, the pressure and temperature variations, p_1 and T_1, do not vary with x while the displacement x_1 varies linearly with the distance x from the closed end. In that case the critical gradient is

$$(\nabla T)_{\text{crit}} = (\gamma - 1)/(\beta x)$$

where γ is the ratio of the specific heats. We can, in any case, write the excess temperature as

$$\delta T = T_1(\Gamma - 1)$$

The next step in the cycle is a rapid outward motion of the parcel of gas, moving it back by x_1 to its original position, where it again sits long enough to equilibrate with the plate (figures 3c and 3d). The adiabatic expansion reduces the pressure by p_1 and the temperature by T_1, and the isobaric cooling (for $\Gamma > 1$) then restores the initial condition of the parcel.

The net effect is to transfer heat along the plate—from right to left or left to right, depending on whether the parameter Γ is larger or smaller than unity. During each cycle the parcel transports an amount of heat

$$q = v\rho c_p \delta T = v\rho c_p T_1(\Gamma - 1)$$

over the distance T_1. As it absorbs heat the entropy of the parcel increases by q/T; half a cycle later its entropy decreases by the same amount. Thus over each cycle an entropy $2q/T$ is transported from right to left. (This argument ignores the relatively small entropy, of order $(T_1/T)(q/T)$, produced by irreversible thermal conduction in one cycle.)

In an actual engine the heat transported to the left by this parcel is in turn picked up by another parcel during the next cycle of acoustic oscillations, and so forth. Figure 4 shows this process. A single short plate of length Δx is connected to hot and cold reservoirs at T_H and T_C, producing a gradient ∇T. The plate is in a long tube filled with gas undergoing acoustic oscillations. The plate is a distance x from the closed end of the tube and the oscillations have a wavelength λ, with

$$\lambda/2\pi \gg x \gg \Delta x$$

The amplitudes of the dynamic pressure and velocity are then essentially constant over the plates.

In a gas engine it is easy to make the effective heat capacity per unit of boundary area of the plates much larger than that of the gas, so the plate temperature varies only slowly, not at the acoustic frequency. Not all the gas is in thermal contact with the plates—effectively, only that gas within a thermal penetration depth δ_κ of the plates. This is the distance, $(2\kappa/\omega)^{1/2}$, over which heat diffuses in the time

Acoustic engine. As parcels of gas oscillate from **a**, **b**, **c**, ... to **a'**, **b'**, **c'** ... and back, heat is transported from the hot end at T_H to the cold end at T_C, but some of the energy absorbed from the hot end is converted to work, which drives the acoustic oscillations. Gas within a thermal diffusion length δ_κ of the plate is effectively in thermal contact with the plate. Figure 4

ω^{-1}, where ω is the angular frequency of the sound and κ is the thermal diffusivity of the gas. If the spacing between the plates is substantially greater than δ_κ, then the plates respond equally and independently, and it suffices to consider one plate only. Further, if the viscous penetration depth is much less than δ_κ, the gas moves spatially uniformly with respect to the adjacent plate, so far as thermal interaction is concerned. Moreover, if the longitudinal heat conduction in the gas and plate are small enough, the only diffusive heat transfer of interest is lateral.

Because thermal diffusion limits the amount of gas on "speaking terms" with the plate, we can confine our attention to a volume of gas that extends a distance δ_κ from the plate, indicated by the colored region in figure 4; the dimensions of this volume are Δx, δ_κ and Π, the perimeter of the plate in the direction normal to x.

As in our discussion of figure 3, we will assume a stepped, articulated cycle rather than truly sinusoidal oscillations. Then, during each cycle of acoustic oscillation, parcels of gas, labeled **a**, **b**, **c**, ... in figure 4, move rapidly inward, each a distance x_1, increasing in pressure by p_1 and increasing in temperature adiabatically by T_1. The parcels wait there—at positions **a'**, **b'**, **c'**, ...—briefly, allowing heat to be exchanged laterally with the adjacent plate; they then move rapidly outward a distance x_1, decreasing the pressure by p_1 and cooling adiabatically in temperature by T_1, and again wait there to exchange heat laterally, returning to their initial condition. In a full cycle each parcel carries an amount of heat q and entropy $2q/T$ to the left. Heat rejected by **a**, for example, is picked up by **c'** half a cycle later; that picked up by **a'** was rejected by **b** half a cycle earlier, and so forth. Heat is thus simply shuttled along the plate; this is the surface heat pumping described[8] by William Gifford and Ralph Longsworth for their "pulse tube." But at, for example, the right end, parcel **d** breaks contact on moving to **d'**, so that it has the plate temperature at **d**, a hotter temperature by T_1 at **d'**, and no heat transfer to the plate. The heat absorbed by **b'** from the plate at T_H thus is not compensated, and net heat q flows from T_H every cycle. This heat is then shuttled down the plate and, by similar reasoning, heat q is rejected to T_C. The broken symmetry at the ends of the plate leads to the heat transfer from T_H to T_C.

The effective volume rate of flow of the gas is $\Pi\delta_\kappa u$, where u is the flow speed of the gas. The heat per unit volume transported down the plate is q/v, which we computed above. The hydrodynamic heat flow from T_H to T_C is thus

$$\dot{Q} = \alpha \Pi \delta_\kappa \rho c_p T_1 u(\Gamma - 1)$$

where α is a constant of order unity. This result is independent of the length Δx of the plate, as is clear physically from the heat-shuttling picture shown in figure 4. When Γ is less than 1, $\dot{Q}$ is negative and heat is pumped hydrodynamically toward the closed end, as we expect from the results shown in figure 2. (The parameter Γ is zero at the beginning of that run.)

During each expansion at the higher temperature, each parcel of the gas does, as we have mentioned, an amount of work w/v per unit volume; the work per cycle done by the entire volume of the gas is thus $\Pi\Delta x\delta_\kappa p_1\beta T_1(\Gamma - 1)$. Because the tube is closed at $x = 0$, the pressure increase p_1 can be computed from the adiabatic compressibility k_s of the gas:

$$k_s p_1 = v_1/v = x_1/x$$

The rate at which the gas does work is, roughly, the work per cycle times the frequency ω, so that

$$\dot{W} = \alpha' \Pi \delta_\kappa (\beta/k_s) T_1 u(\Gamma - 1)(\Delta x/x)$$

Here α' is another constant of order unity, and we have used $u = \omega x_1$. For a fluid of zero viscosity, both α and α' are ¼. For a nonviscous fluid, the efficiency of this engine is

$$\begin{aligned}\eta &= \dot{W}/\dot{Q} \\ &= (\beta/k_s \rho c_p)(\Delta x/x)\end{aligned}$$

A thermodynamic result relates the adiabatic compressibility and the thermal expansion coefficient of the fluid:

$$\beta/k_s \rho c_p = (\gamma - 1)/T\beta$$

The efficiency for the simple acoustic engine using a nonviscous fluid is then

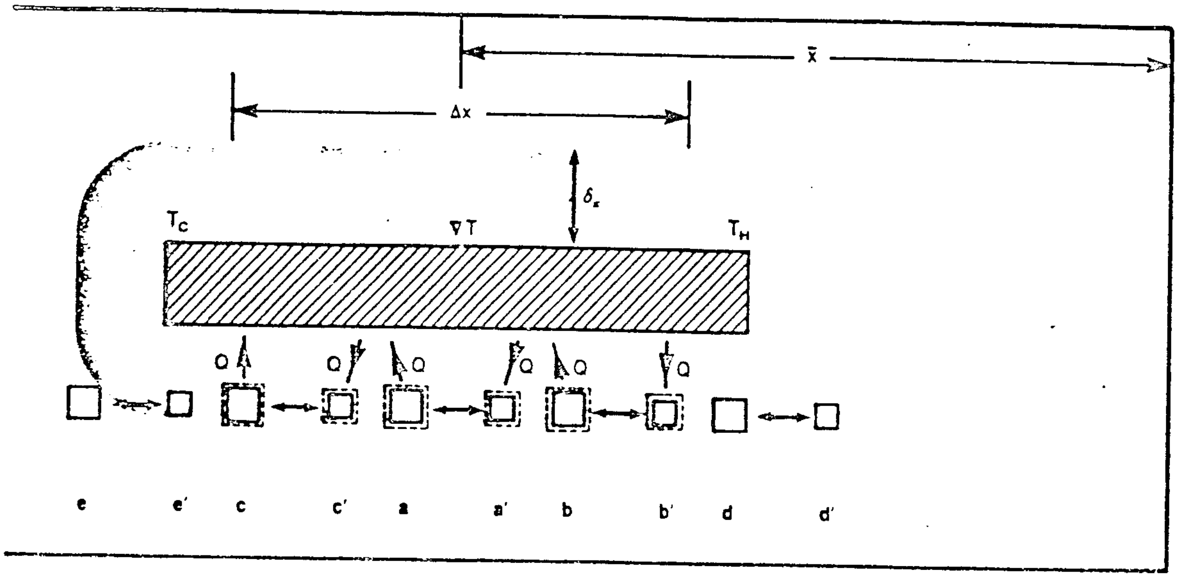
$$\begin{aligned}\eta &= [(\gamma - 1)/T\beta](\Delta x/x) \\ &= \Delta x(\nabla T)_{\text{crit}}/T\end{aligned}$$

Remarkably, this efficiency depends only on fluid parameters, geometry, and configuration. Yet it is always less than the Carnot efficiency. The Carnot efficiency is to be expected only as the heat transfer rate approaches zero, that is, as (∇T) approaches $(\nabla T)_{\text{crit}}$. In this limit, $(\Delta x)(\nabla T)_{\text{crit}}$ becomes ΔT, the temperature difference along the plate, and the limiting efficiency is $\Delta T/T$, the Carnot efficiency.

Figure 4 also illustrates that this natural engine is in fact a large number of engines in series. While T_1, the adiabatic temperature change of a given parcel, may be small, the total range $T_H - T_C$ spanned by the engine can be large, as the number of parcels in series, $\Delta x/x_1$, can be large.

Some thermoacoustic engines

Thermoacoustic natural engines may eventually have some practical use because of their great simplicity. Figure 5 shows schematically the forms of three engines, two of which have actually been realized in practice. On



the left (figure 5a) is a cryocooler in a form developed by Thomas Hofler at Los Alamos. It consists of an acoustic driver D (in the form of a loudspeaker), replacing the closed end of the semi-infinite tube of our simple example; a heat exchanger H, to make contact with a hot reservoir; a stack of plates S; a heat exchanger C, to make contact with a cold reservoir; and a long tube terminated by a bulb R, to serve as the remainder of the acoustic resonator.

For monatomic gases of zero viscosity the critical temperature gradient $(\nabla T)_{crit}$ is $2T/3x$, where x is the distance to a pressure antinode. It is not possible to "pump up" a temperature gradient exceeding this value, so to obtain a large temperature ratio T_H/T_C across the stack of plates, one must make x at the "hot" (ambient temperature) end as small as possible. Moreover, the cold end of the stack and the remainder of the acoustic resonator should be thermally isolated. Hofler's model incorporates these features. It has produced a temperature of −78 °C at the cold end when the temperature at H was 23 °C under no load. The temperature at the cold end rises to −50 °C when a heat load of 2.5 watts is applied to it.

The heat-operated cooler (figure 5b and cover) is the most aesthetically satisfactory of our thermoacoustic engines. It has two stacks of plates, S_1 and S_2; and there are four sets of heat-exchange plates, H, A_1, A_2 and C, oriented at right angles to S_1 and S_2. The top set, H, is maintained at a high temperature, say, 450 °C; A_1 and A_2 are maintained near ambient temperature, while C and the resonator R are at a subambient temperature. The temperature T_H is large enough that stack S_1 produces work, which in turn drives stack S_2 as a heat pump. Thus there are acoustically stimulated heat flows from both ends toward the middle. The "cold" temperature cannot be pushed far below ambient, typically not much below 0 °C, as S_2 is some distance from the closed end; the critical temperature gradient is thus low and viscous losses are higher than in the cryocooler.

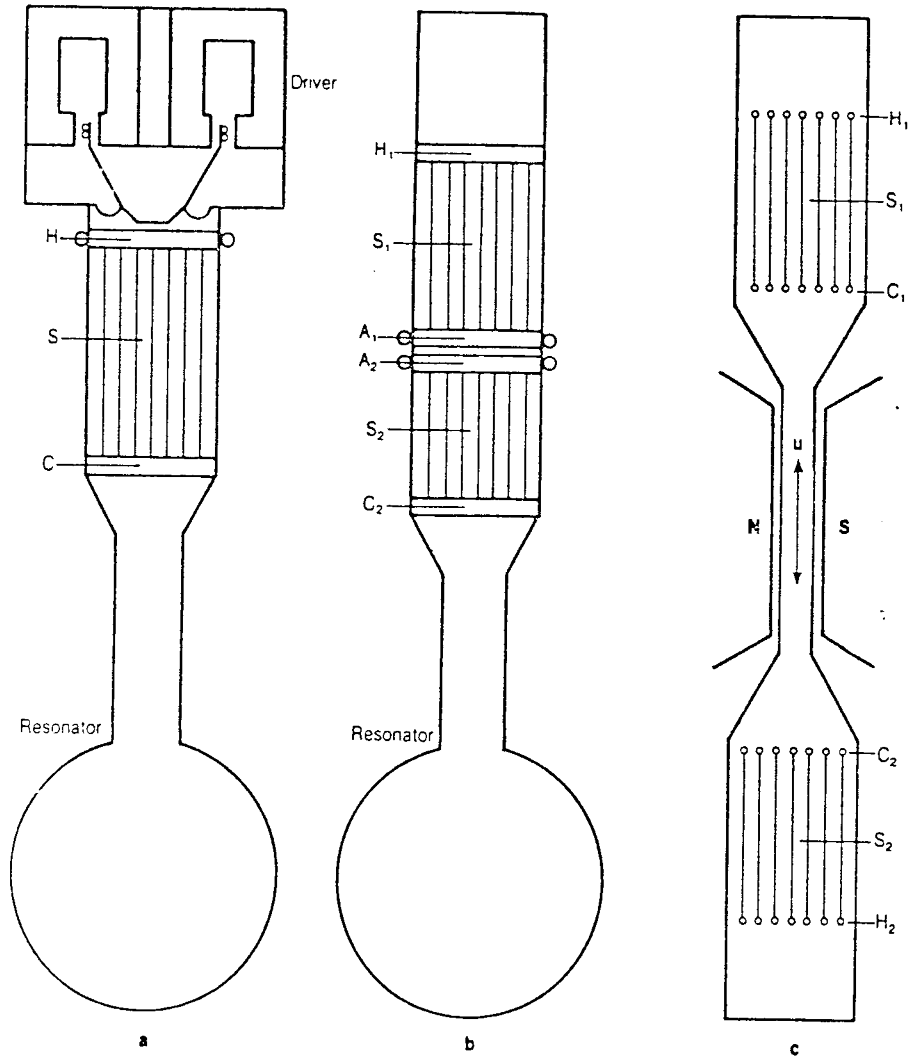
Figure 5c is a schematic drawing of a proposed liquid-sodium thermoacoustic prime mover currently being worked on by Gregory Swift and Albert Migliori at Los Alamos. This engine was motivated by John F. J. Malone's pioneering work[10] on liquids working in heat engines. Malone suggested that liquid sodium might be a good thermodynamic working substance. Such an engine may also meet the need for a simple and reliable prime mover for electrical power generation in space. The proposed engine is a half-wave acoustic resonator that has two symmetrically located stacks of plates, S_1 and S_2; hot heat exchangers, H_1 and H_2; and "cold" heat exchangers, C_1 and C_2. There is a velocity antinode at the center, where electrical power is extracted magnetohydrodynamically, utilizing the electrical conductivity of the sodium. Liquid sodium has a very low ratio of viscous to thermal penetration depths, because the thermal conductivity is greatly enhanced by electronic heat conduction. The engine is therefore a rather efficient prime mover and is characterized by very high oscillating pressures. We have calculated that if the engine operates between 1000 K and 400 K it will generate about 60 W/cm² of acoustic power with an efficiency about ⅓ of the Carnot efficiency. The performance is expected to be substantially better at higher operating temperatures.

Variable stars

The concepts of natural engines are helpful in understanding variable stars, such as the famous Cepheid variables. Earlier in this century, astrophysicists believed that a time-modulated heat source, as one often finds in terrestrial engines, was the source of the variability. Present-day models attribute the observed variations in surface motions and total light from a variable star to one or more of several "valving" mechanisms,[11] some of which we will describe here.

Periods of variable stars range from seconds, for white dwarf stars, to years, for the gigantic Mira variables. If the star evolves to a pulsationally unstable state, it will start to oscillate with a growing amplitude, typically increasing its pulsation kinetic energy by perhaps one percent each period. (These growth rates vary widely, however, from case to case.) At the limiting amplitude, the energy of the stellar pulsation is typically only one billionth of the internal heat energy in the star.

The most important mechanism for causing stars to pulsate, that is, to convert some of the luminosity—outward energy flux—into motion, is the kappa mechanism. This is named after the Greek letter κ, which denotes the opacity of the stellar material. In a normal star, energy generated by nuclear reactions in the central regions flows out through the envelope by photon transfer. This photon transfer is not very efficient, because each photon typically goes only one centimeter before it is absorbed by an atom or scattered by a free electron. That is, the photons *diffuse* through the stellar material instead of simply propagating out from the core of the star. A rather simple theory can very accurately describe this diffusive transport of radiation, and one can calculate a mean absorption. As a parcel of stellar material moves through its trajectory, its mean absorption varies cyclically, blocking and releasing photons through the stellar envelope and causing stellar pulsations under appropri-



Examples of thermoacoustic engines: **a** cryocooler, **b** heat-operated cooler, **c** liquid-sodium prime mover. In these sketches, S are stacks of plates (as in figure 2) connected to heat exchangers at hot (H), cold (C) and ambient (A) temperatures. The cryocooler is driven by a loudspeaker. The heat-operated cooler is also shown on the cover, with a torch supplying energy instead of the usual heat-exchange fluid. The prime mover has been proposed as a reliable and comparatively efficient electric-power generator for space vehicles. Figure 5

ate circumstances. The variation of opacity is the means by which thermodynamic symmetry is broken.

In the model for stellar pulsations we consider first, the kappa mechanism occurs simultaneously with another process called the gamma mechanism (after the Greek letter γ that denotes the ratio of specific heats at constant pressure and volume).

The graph in figure 6 shows how κ varies with temperature at three densities ρ for typical stellar material. At intermediate temperature, perhaps 10^4 K, the opacity is high, owing to absorption (as photoelectric effect) and scattering. But as the temperature rises, the stellar fluid becomes more fully ionized, so the number of possible photon-absorption processes decreases; the opacity thus decreases. Of course, increasing the density at constant temperature increases the opacity.

Because the gas in a star behaves nearly like an ideal gas, an appropriate equation of state is

$$\frac{dT}{T} = (\gamma - 1)\frac{d\rho}{\rho} + \frac{ds}{c_v}$$

or alternatively

$$\frac{dp}{p} = \gamma\frac{d\rho}{\rho} + \frac{ds}{c_v}$$

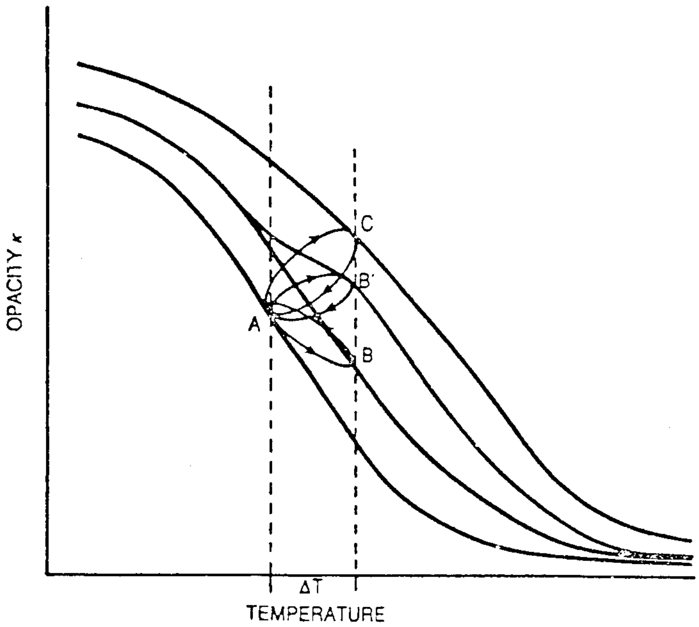
For a completely ionized gas with little entrapped radiation, the value of $\gamma - 1$ is near the ideal-gas value of $2/3$; in that case, the gas is comparatively incompressible, or "hard." However, at lower temperatures $\gamma - 1$ can be rather small—perhaps 0.1—so that the gas is easily compressible, or "soft." The reason is that only some of the internal energy derived from work done on the gas goes to an increase in the kinetic energy of the atoms and electrons (and thus an increase in temperature and pressure); much of the energy goes to increasing the ionization of the gas.

Whether the stellar fluid is "hard" or "soft" has a direct effect on the changes in the opacity κ during a cycle, and hence directly affects how the thermodynamic symmetry is broken. For example, suppose a parcel of stellar fluid initially at A in figure 6 is compressed to increase its temperature by ΔT. In a hard material, where $\gamma - 1$ is relatively large, the final state might be that indicated at B. Because the opacity is less than at A, there is no tendency for oscillations to start. But for a softer material, where $\gamma - 1$ is relatively smaller, the density change is larger, the final density and the opacity are those at C. Thus the γ effect acts within the dynamics of the stellar processes to enhance or diminish changes in the opacity, which determine the thermal interaction of a parcel of stellar fluid with the radiation flow.

Suppose a parcel of stellar fluid follows a cycle between points A and C in figure 6, with extremes of density ρ_1 and ρ_3. During its compression from A to C, the parcel's opacity is progressively increasing and the radiation flux adds heat to the parcel as the pressure increases; the temperature thus continues to rise even after the density has maximized. During expansion from C to A, the opacity drops, heat is released more readily, and the temperature continues to drop, even after the density has minimized. The effect of such processes accumulates for other parcels of stellar fluid within the region of the star where the values of κ and γ vary appropriately.

In a radially pulsating star, where the frequency is one of the normal radial compressional frequencies of the stellar object, the thermodynamically active region extends from near the stellar surface, where the mass density and heat capacity are becoming small, to well within the star, where ionization is more complete and the stellar fluid becomes too "hard" to drive pulsations. The region of the star that acts to drive the pulsations—that is, the region in which the phasing of thermal contact with the pressure variations is favorable—can be thought of as analogous to the prime-mover plates in an acoustic engine (such as the upper group of plates in figure 5b). In other regions of the star—for example, where the stellar fluid is hard—the phasing of the thermal contact with pressure variations may be unfavorable. This effect is called radiation damping or leaking. These regions absorb the work generated by the stellar prime mover, thus limiting the amplitude of the pulsations. The work-absorbing part of this pulsating star is analogous to the lower group of plates in figure 5b.

It is also possible to have a "pure κ



Stellar oscillations. The solid curves show opacity as a function of temperature for stellar material at three different densities. Cyclic expansions and contractions of relatively "hard" material between states A and B do not spontaneously grow into stellar pulsations. For a "soft" material, or one whose composition is such that the opacity follows the colored curve, the oscillations will grow in time until they are damped by some other mechanism. Figure 6

effect" at higher temperatures, where the ionization is so complete that the stellar material is never soft. For a pure κ effect to work, the opacity plotted as a function of temperature at constant density may appear to have a "bump" on it, as shown by the colored curve in figure 6. Such a bump might arise from K-shell ionizations of helium, carbon and oxygen, whose abundances in stars sometimes are large enough to lead to a noticeable increase of opacity on adiabatic compression, and hence to a tendency toward stellar pulsations, as indicated by the fact that a compression by ΔT leads to a larger opacity at B′ rather than the smaller value at B.

The first prime-mover mechanism discussed above operates near the surface of a variable star. Temperatures are generally less than a million kelvin there, and often the driving is in layers as cool as 10 000 K. The damping region in stars ranges in temperature between 50 000 K and a few million kelvin, depending on such circumstances as the mass and composition of the star.

Another pulsation mechanism involves a driving mechanism deep in the star; the damping mechanism is at the same location in the star. The mechanism is analogous to the process depicted in figure 3. In the discussion of that figure, we distinguished between work-producing and work-absorbing engines, that is, between temperature gradients in the plate that exceed the critical temperature gradient and those that are smaller. The critical temperature gradient is the ratio of the adiabatic temperature change of the fluid parcel itself to the distance it moves during that temperature change. In discussing the motion of gas in stars, just as in discussing the motion of fluids in meteorology or oceanography,[12] it is instructive to consider a parcel of fluid moving with respect to a fixed "surrounding" fluid. The pressure on the moving parcel is changing; in an adiabatic process its temperature is also changing. Let the trajectory of temperature versus position be called the "adiabatic." In natural engine language the slope of this trajectory would be the "critical temperature gradient." If the temperature gradient of the surrounding fluid exceeds the critical gradient it is called "superadiabatic." The oscillatory instability involves motions of convective eddies in a star in a superadiabatic surrounding fluid where the amplitude of the convective motions is restrained by a composition gradient. The need for the surrounding fluid to be superadiabatic is just the same as the requirement that the temperature gradient in the walls in figure 3 exceed the critical gradient, if work is to be produced. The analogous process in the atmosphere or ocean leads to the Brunt–Väisälä instability.

Normal early stellar evolution converts hydrogen with low atomic weight to helium, carbon, oxygen and other heavier elements. Many stars more massive than our Sun develop an atomic-weight gradient region, typically when the temperature is between 10 and 50 or more million kelvins, with the species of higher atomic weight closer to the center of the star. Now imagine that a convective eddy—in pressure equilibrium with its surroundings but with a composition characteristic of a greater depth within the star—moves a parcel up, into a region with more hydrogen and less helium, where the parcel is therefore denser than its surroundings. This will eventually stop the upward motion, and the atomic weight gradient will make the eddy sink and return to its mean position. But because of the superadiabatic surroundings, the parcel loses energy to the surroundings during its upward excursion, and when it returns to its original depth it is cooler and denser than its surroundings. It thus continues sinking to deeper levels, where—again because of the superadiabatic gradient—it gains heat from the surroundings. If these deep oscillations are not stabilized by radiative damping in the regions nearer the surface, the star will pulsate in one—or perhaps many—nonradial modes. This destabilization is called the Kato mechanism.[13] Whether or not the Kato mechanism actually produces observable variable stars is not yet known.

References

1. K. T. Feldman Jr, J. Sound Vib. **7**, 71 (1968).
2. K. T. Feldman Jr, J. Sound Vib. **7**, 83 (1968).
3. Y. Yazaki, A. Tominaga, Y. Narahara, J. Low Temp. Phys. **41**, 45 (1980).
4. J. Wheatley, D. S. Buchanan, G. W. Swift, A. Migliori, T. Hofler, submitted to Proc. Natl. Acad. Sci. USA.
5. J. W. Strutt, *Theory of Sound*, Dover, New York (1945), section 322.
6. J. Wheatley, T. Hofler, G. W. Swift, A. Migliori, J. Acoust. Soc. Am. **74**, 153 (1983).
7. P. Merkli, H. Thomann, J. Fluid Mech. **70**, 161 (1975).
8. W. E. Gifford, R. C. Longsworth, Intl. Adv. Cryo. Eng. **11**, 171 (1966).
9. N. Rott, Adv. Appl. Mech. **20**, 135 (1980); N. Rott, J. Fluid Mech. **145**, 1 (1984).
10. J. F. J. Malone, J. R. Soc. Arts (London) **79**, 679 (1931).
11. J. P. Cox, *Theory of Stellar Pulsations*, Princeton U.P., Princeton, N.J. (1980); A. N. Cox, Publ. Astron. Soc. Pac., a review in preparation (1985).
12. C. Normand, Y. Pomeau, M. G. Velarde, Rev. Mod. Phys. **49**, 590 (1977).
13. S. Kato, Publ. Astron. Soc. Jpn. **18**, 374 (1966).

Ivars Peterson reports on a meeting of the Acoustical Society of America

A sound way to generate electricity

SCI NEWS 31 MAY 86

Converting heat energy into electricity is nothing new. That's what happens in a coal-fired power plant, for instance. But transforming heat energy first into sound energy and then into electrical power is somewhat unconventional. This latter process is the basis for a liquid-metal acoustic heat engine now being developed at the Los Alamos (N.M.) National Laboratory (LANL).

Although the acoustic engine isn't as efficient as other heat engines, such as steam turbines, it has the advantage of having no moving mechanical parts and hence a high reliability, says LANL's Greg Swift. Such engines could be used to provide electrical power for satellites and other applications where reliability is extremely important.

The prototype liquid-metal acoustic heat engine consists of a 1-meter-long metal tube, closed at both ends and filled with liquid sodium. A stack of thin parallel plates, made from molybdenum, sits inside the cylinder. Two sets of tiny tubes carry fluids that keep one end of the plates at 125°C and the other at 700°C.

"Sound is generated by the temperature difference across that stack," says Swift, who with Al Migliori designed the engine. "It appears spontaneously," he says. "The liquid sodium starts to sing." In other words, the liquid begins to oscillate. These oscillations in the presence of a magnetic field at the tube's midpoint allow an electrical current to be generated.

So far, in two separate projects, the researchers have confirmed that they can, indeed, produce sound from heat and that their magnetohydrodynamics generator does convert acoustic waves into electrical energy. Eventually, the two steps will be combined into a single working generator. "Nothing we have learned has discouraged us yet," says Swift.

The sodium engine is just one extension of earlier work on acoustic heat engines (SN:12/4/82,p.358). Tom Hofler, a graduate student from the University of California at San Diego and now working at LANL, is developing a loudspeaker-driven refrigerator. In this case, electrical energy is converted into acoustic waves (as in a loudspeaker) and these sound waves produce a cooling effect.

OMNI
march 1989 p.37

MUSICAL REFRIGERATION

When NASA found out it couldn't operate conventional refrigeration units in zero gravity, the agency thought it was stuck. But that was before a team of scientists at Los Alamos National Laboratory developed a refrigerator that runs on, of all things, sound. Now NASA may have a prototype refrigeration unit on a shuttle sometime this year. "We have a loudspeaker-driven refrigerator with an ordinary hi-fi midrange for power that sends sound-wave oscillations through compressed helium in a tube with heat exchangers," says Los Alamos physicist Greg Swift of the invention. The vibrations across a temperature gradient cause the helium to expand and contract as much as 1,000 times every second. "This breathing action pumps heat out of a refrigerated box much like your Frigidaire at home," notes Swift.

But why use such a bizarre system? "NASA," Swift explains, "found that you can't run conventional refrigeration units in space because they require gravity to work. The acoustical engine doesn't, plus there are no moving parts in an acoustical heat exchanger, so it'll never break down."

Swift and other researchers are also working on heat-driven engines that function in much the same way. The heart of the system is a metal tube, with heat exchangers and parallel metal plates at one end. When heat or sound waves pass through liquid sodium in the pipe, they create heat and magnetic energy, which can be transformed into electricity. The only moving part is the liquid sodium. Swift, who predicts that the practical application of these engines is about ten years away, admits that liquid sodium is not very easy to work with; but then again, it never needs oiling, won't wear out, and can't break. And someday these engines might power our large industrial plants. "I don't see why we couldn't make larger acoustic heat engines," Swift says. "The bigger we make them, the more efficient they'll be."—Bob Mangino

The sounds of science: Los Alamos scientists give NASA a hand by designing a refrigerator that's powered by sound.

"Conscience gets a lot of credit that belongs to cold feet."

—Anonymous

"We are absolutely certain only about things we do not understand."

—Eric Hoffer

REFRIGERATOR MAKES CHILLING SOUNDS

By DAVID SCOTT

Here's the ultimate in cool sounds: a refrigerator that's powered by a loudspeaker and uses no chlorofluorocarbons (CFCs) for cooling.

Researchers at the Naval Postgraduate School in Monterey, Calif., have developed a "cryocooler," a new breed of thermoacoustic engine that cools using sound waves and a mixture of helium and argon instead of the conventional compressor and ozone-destroying CFC refrigerant gas.

When current is applied to a loudspeaker, it begins to vibrate. As the diaphragm pulses outward, nearby molecules compress and heat up. When the diaphragm moves back in, the molecules expand and cool down. In most cases—say, in a room or an organ pipe—the heating and cooling cancel each other out.

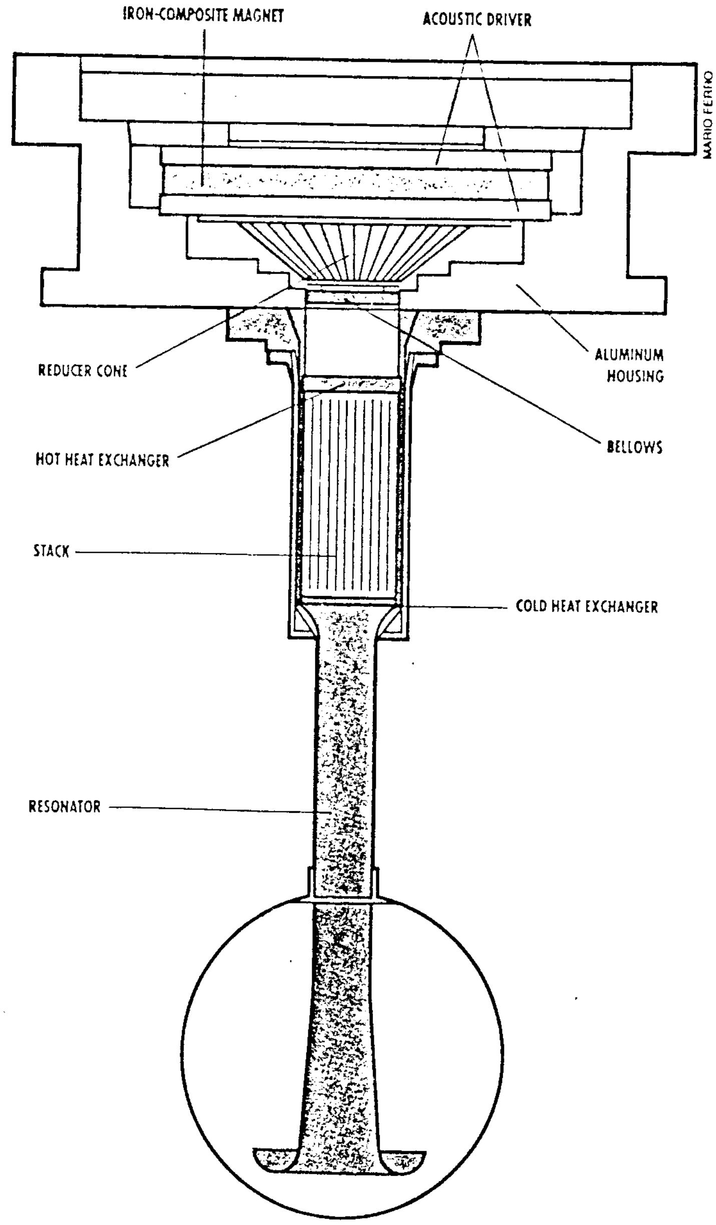
The thermoacoustic effect of heat converting to sound was first observed centuries ago by glass blowers, who noticed that the tube attached to a hot expanded-glass bowl would tend to cool and begin "singing." The first demonstration of the reverse of the process—sound used to pump heat for cooling—was in 1982. Now physics professor Steven Garrett and his colleagues at the school, Thomas Hofler, Jay Adeff, and Glenn Harrell, have figured out a way to harness this cycle into an efficient system for refrigeration. Here's how it works:

A small loudspeaker, just like the ones in your home or car stereo system, sits atop a rigid chamber of pressurized helium and argon gas. Plastic baffles—the crucial heat-trapping component—fill the upper part of the chamber. When current is applied to the speaker, sound waves compress and heat up nearby gas molecules. These ram into the plastic, transfer some of their heat, and cool down a bit. After expanding, the molecules end up with less energy and are cooler than when they began the cycle. Heat exchangers inside the baffles pull heat away from the plastic, and the gas inside the chamber gets cold, eventually leaving the bottom of the tube freezing cold.

"Basically, what you've got here is a bucket brigade, with molecules passing heat from the gas in the chamber to the plastic," says Garrett. By passing a loop of something like liquid antifreeze over the chamber, the cold can be circulated around the walls of a storage compartment in a refrigerator.

At 160 decibels, the sound inside the cooler is deafening—10,000 times louder than a Rolling Stones concert. But not a sound can be heard outside the system: Because the cylindrical chamber is rigid enough to withstand the static force from the gas it contains at 10 times atmospheric pressure, the chamber won't vibrate from the far smaller additional pressure exerted by the sound waves.

The researchers have already built a number of working acoustic coolers, some capable of producing temperatures of around minus 100 degrees F. One model is even bound for a ride on the space shuttle as early as January next year. Because they have fewer moving parts than conventional cooling systems, acoustic coolers could be well suited to applications on satellites or space vehicles, where efficient, maintenance-free cooling is crucial, says Garrett. And because the coolers vibrate less than conventional ones, sensitive space telescopes wouldn't jump around as much.



No CFCs required: A thermoacoustic engine in a new refrigerator provides cooling with sound waves and a mixture of helium and argon gas.

Building these cool-sound systems for home use will require some straightforward engineering but shouldn't be terribly difficult, predicts Garrett. The main challenge is designing heat exchangers that will be as efficient as those found in standard refrigerators that use CFCs. The conventional equipment has an advantage because the refrigerant used in the systems also acts as the main heat exchanger. Acoustic coolers, on the other hand, require separate stages: cooling the helium mixture to remove heat from a liquid or gas, which in turn circulates through the appliance. The researchers will also work to adjust the size of the cooling system and the frequency of the sound to find the most energy-efficient combination.

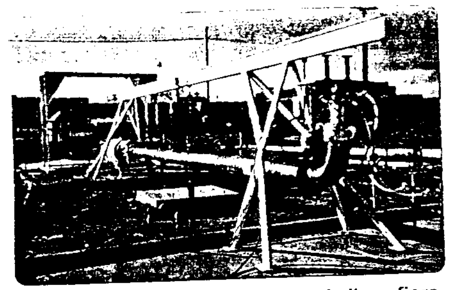
Efforts to adapt the system for home refrigerators and air conditioners are already under way at the Naval Postgraduate School and may be ready for the market in about three years.—*P. J. Skerrett*

Compressed Air Magazine
December 1994 p. 2

MOTIONLESS REFRIGERATION FOR LNG

A new cooperative R&D agreement between the NIST and Cryenco Sciences Inc., of Denver, Colo., aims to apply refrigeration technology developed by NIST and Los Alamos National Laboratory to the task of liquefying natural gas. The name of the invention has more syllables than the invention itself has moving parts. It is a thermoacoustically driven orifice pulse tube refrigerator—TADOPTR, for short. It has no moving parts. The TADOPTR contains tubes of helium gas and is capable of producing a temperature of 112 Kelvin. The helium gas is repeatedly compressed and expanded with sound waves rather than with a mechanical compressor as in most conventional cooling systems. Cryenco has obtained the development license and will upscale the TADOPTR into two versions that will be manufactured, tested, and marketed by Cryenco. The first will liquefy 1900 liters of natural gas per day, while the second will increase that output to 38,000 liters per day. According to NIST, the cost of such liquefaction plants will be economical, with liquefaction taking place on-site.

Popular Mechanics
May 1997 p. 32



Larger versions of these acoustic liquefiers could chill 10,000 gpd of natural gas.

Acoustic Cooler

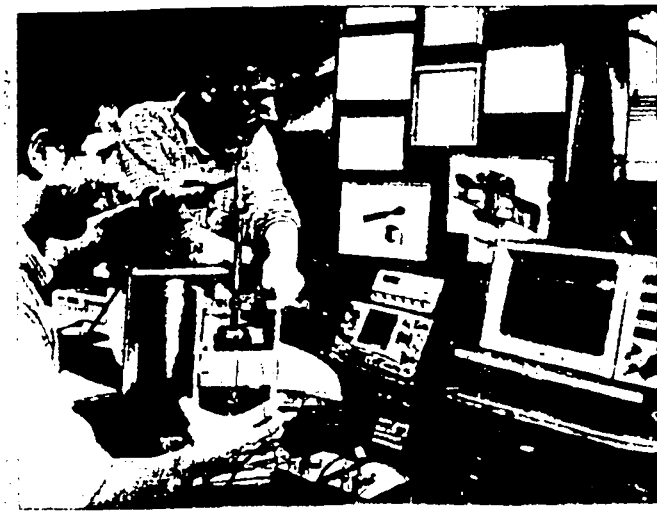
DENVER, CO—It makes good economic sense to transport natural gas as a liquid. But creating the low temperatures needed to do this raises formidable engineering challenges.

A new type of gas liquefaction system, the Tadoptr, could be the answer. Its name is derived from its two principal components, a thermoacoustic driver (TAD) and an orifice pulse tube refrigerator (OPTR).

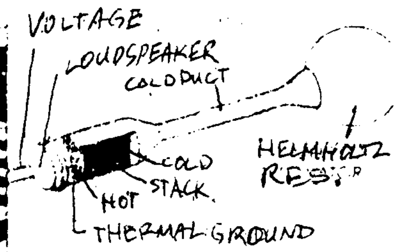
First, a natural-gas burner creates 1400° F temperatures at the TAD end. The heat causes helium inside the tube to vibrate. Then, working much like a Stirling engine, the OPTR uses this acoustic power to extract heat.

Cryenco Sciences, which licensed the technology from Los Alamos National Laboratory, has tested a model that cools gas to minus-240° F. The company hopes to have a 500-gal.-per-day (gpd) system working later this year.

Room-Chilling Sounds



Mechanical engineering doctoral student Minner (left) and Professor Luc Mongeau (right) test the TRD in their Purdue lab



Popular Mechanics
TECH UPDATE
October 1997

"Room Chilling Sounds"

WEST LAFAYETTE, IN—Researchers at Purdue University have found a way to cool air with a loudspeaker.

Their 2-ft.-long thermoacoustic refrigeration device (TRD) consists of a loudspeaker separated from a cavity called a Helmholtz resonator by a tube.

Playing a 300-cycle tone through the speaker causes gas atoms inside the TRD to vibrate, thus setting up pressure fluctuations. "When you compress a gas, it becomes warmer, and when you decompress it, it becomes cooler," explains student Brian Minner, who developed the device with his professor. A porous transfer stack in the center pumps the heat toward the speaker, thus cooling its opposite end.

Acoustic cooling is not a new idea. A large-scale system is currently being used to liquefy natural gas (see Tech Update, page 32, May '97). The device interests appliance makers because of its smaller size, which offers a way to build freon-free refrigerators and air conditioners.

photo caption: Mechanical engineering doctoral student Minner (left) and Professor Luc Mongeau (right) test the TRD in their Purdue lab.

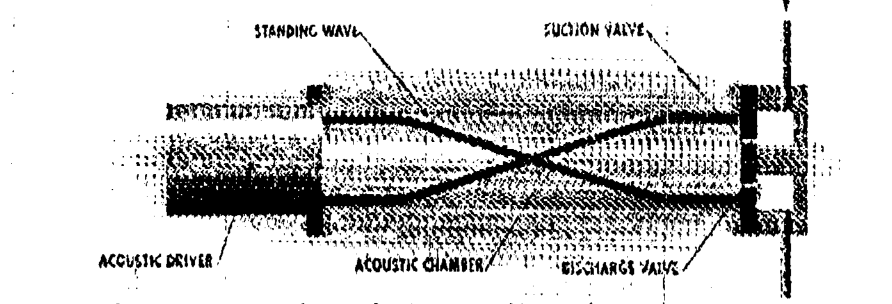
drawing shows: VOLTAGE, COLD DUCT, LOUDSPEAKER, STACK, COLD, THERMAL GROUND, HELMHOLTZ RESONATOR, HOT

A sound idea

Home refrigerators currently account for roughly 8 percent of all electricity consumed in this country, and they rely on ozone-depleting chlorofluorocarbons (CFCs) for cooling. Congress has mandated that 1998 refrigerators must be 30 percent more efficient than 1990 models. In addition, the United States is committed to a 50 percent decrease in CFC use by 1995, with complete elimination by the year 2000.

A new type of refrigerator compressor that uses resonant sound waves will provide energy savings of up to 40 percent and be easily adaptable to new non-CFC refrigerant systems. Developed at Los Alamos National Laboratory in New Mexico by physicist Tim Lucas, who invented it, and Greg Swift, a consultant, the sonic compressor will require no retooling by manufacturers and could be standard equipment in new refrigerators in two or three years. Eventually, it could also be designed for installation in existing refrigerators.

SONIC COMPRESSOR FOR REFRIGERATORS



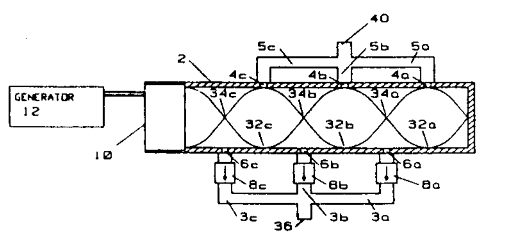
An acoustic driver sets up a sound wave that is converted into usable gas compression by the valves between the chamber and refrigerant lines. Refrigerant circulates to the coils through suction and discharge lines.

In a conventional compressor, the heart of any refrigerator, an oil-lubricated piston-and-cylinder device mechanically compresses refrigerant gas to produce cooling. In the new compressor, an acoustic driver, which functions somewhat like a bass loudspeaker, uses electrical power to set up resonant sound waves with pressure amplitudes up to 100 pounds per square inch. The sound waves then compress the refrigerant when they pass through a pair of high speed, 340-cycles-per-second, one-way valves. Lucas has formed Sonic Compressor Systems of Richmond, Va., to commercialize the new device.

Thermoacoustics, another sound-cooling technology, from the Naval Postgraduate School in Monterey, Calif., is also moving ahead ["Best of What's New," Dec. '91; "Home Newsfront," July '91]. Cool Sound Industries in Port St. Lucie, Fla., has exclusive license for the thermoacoustic air-conditioning, and General Electric is building home refrigerators in Monterey. The technology could be commercially available in three years.
—*Ray Nelson*

June 14, 1991 US Patent Gazette

5,020,977
STANDING WAVE COMPRESSOR
Timothy S. Lucas, 4614 River Mill Ct., Glen Allen, Va. 23060
Continuation-in-part of Ser. No. 256,322, Oct. 11, 1988, abandoned. This application Jul. 12, 1989, Ser. No. 380,719
Int. Cl.[5] F04B *35/00*
U.S. Cl. 417—322 **23 Claims**

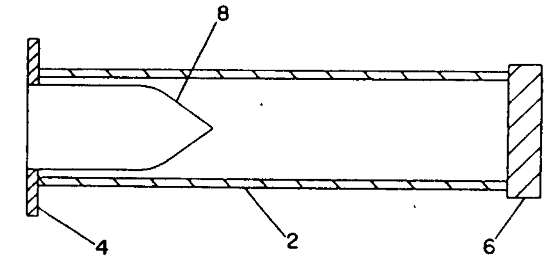


1. A refrigerant compressor comprising:
(a) a chamber for receiving a gaseous refrigerant to be compressed and discharged;
(b) a driving means for establishing a traveling acoustic wave in said gaseous refrigerant in said chamber;
(c) means including said driving means and said chamber for converting said traveling acoustic wave to a standing acoustic wave in said gaseous refrigerant in said chamber having one or more pressure nodes and one or more pressure antinodes therein;
(d) one or more input ports in said chamber;
(e) one or more output ports in said chamber.

June 14, 1994

5,319,938
ACOUSTIC RESONATOR HAVING MODE-ALIGNMENT-CANCELED HARMONICS
Timothy S. Lucas, Lafayette, Ind., assignor to Macrosonix Corp., Glen Allen, Va.
Filed May 11, 1992, Ser. No. 881,339
Int. Cl.[5] F25B *9/00*
U.S. Cl. 62—6 **22 Claims**

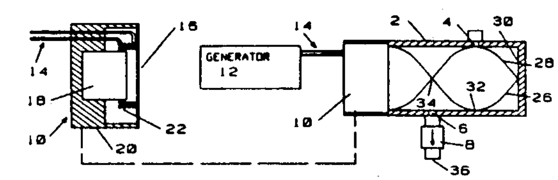
1. An acoustic resonator comprising a chamber containing a fluid, said chamber having a geometry which produces destructive self-interference of at least one harmonic in said fluid



to avoid shock wave formation at peak acoustic pressure amplitudes greater than 10% of mean pressure.

OCTOBER 25, 1994

5,357,757
COMPRESSION-EVAPORATION COOLING SYSTEM HAVING STANDING WAVE COMPRESSOR
Timothy S. Lucas, Glen Allen, Va., assignor to Macrosonix Corp., Glen Allen, Va.
Continuation of Ser. No. 943,534, Sep. 11, 1992, abandoned, which is a continuation of Ser. No. 665,316, Mar. 6, 1991, Pat. No. 5,167,124, which is a division of Ser. No. 380,719, Jul. 12, 1989, Pat. No. 5,020,977, which is a continuation-in-part of Ser. No. 256,322, Oct. 11, 1988, abandoned. This application Oct. 26, 1993, Ser. No. 141,697
Int. Cl.[5] F25B *9/00*
U.S. Cl. 62—6 **3 Claims**



1. A method of compressing a fluid compression-evaporation refrigerant comprising the steps of:
introducing a fluid compression-evaporation refrigerant to be compressed into a chamber having at least one inlet and at least one outlet; and
establishing a standing wave in the fluid compression-evaporation refrigerant in the chamber, so that the fluid refrigerant is compressed.

Cool Sounds

An acoustic compressor pumps up the volume on refrigeration

After years spent developing whisper-quiet refrigerators, appliance makers may turn to noise to cool the next generation of iceboxes. A physicist-turned-entrepreneur has developed a lubricant-free sonic compressor that promises to be more energy efficient than standard compressors and can use refrigerants not based on ozone-depleting chlorofluorocarbons (CFCs).

Timothy S. Lucas, president of Sonic Compressors Systems in Glen Allen, Va., believes his compressor may prove to be the tonic the appliance industry needs to meet the upcoming stringent CFC regulations and energy efficiency requirements. Lucas says his start-up company has built working prototypes and is negotiating a contract with a major refrigerator maker. He hopes to demonstrate his device in a domestic refrigerator in early 1993.

What makes the instrument promising is that it is a drop-in replacement for conventional compressors and so will not require retooling or other expensive procedures. In addition, "we should be able to make the compressors at a comparable price," Lucas says. Coupled with a projected improvement in energy efficiency of 30 to 40 percent over existing compressors, manufacturers may save on more expensive techniques for efficiency, such as increasing insulation or adding larger heat exchangers and more powerful fans.

Standard compressors in refrigerators generally rely on pistons or rotors to compress the cooling gas. Because refrigerators are expected to function trouble free for 15 to 20 years, the compressors must be lubricated to prevent excessive wear. And the refrigerant and the lubricant must be compatible. The most ideal mix has been mineral oils and CFC 12, which is scheduled to be phased out over the next few years, possibly by 1996. As a replacement, the chemical industry is pushing hydrofluorocarbon (HFC) 134a, a substitute that developers claim has no known effect on the ozone layer and should produce only a minimal greenhouse effect.

Some refrigerator engineers worry that lubricants compatible with the new refrigerant may not be developed in time. The appliance industry has been evaluating ester-based oils, but tests so far have not been encouraging. "The lubrication properties are not as good as those of mineral oils," says Carl Offutt, the general manager of engineering at the Whirlpool Corporation. Furthermore, "funny things start happening when you add other compounds to improve the lubricant's effectiveness," he remarks. Over time the ester oils react with HFC 134a and other oils used to manufacture components to form a waxy residue that plugs up the refrigerator tubes. The oils, too, may react with water to corrode the internal parts, so that refrigerators would have to be made in a moisture-free environment.

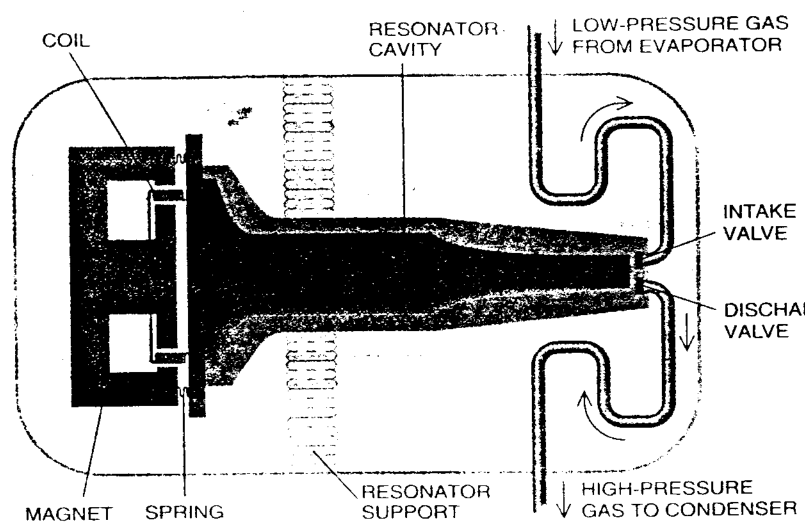
In contrast, Lucas's compressor needs no lubrication, and that means "throwing away oil restraints" in finding an environmentally friendly refrigerant, Lucas says. The sonic compressor is simply an oddly shaped tube that acts as a resonance cavity for the refrigerant. The entire resonator moves back and forth about 50 microns along its cylindrical axis at about 340 hertz. The oscillation creates a standing wave in the cavity. Because the cavity is designed so that the standing wave reinforces itself, the pressure changes achieved in the tube are large. In terms of sound pressure, the amplitude is about 200 decibels, but the compressor is nonetheless quiet because the mass of the tube prevents any sound from escaping. A valve vents the pressurized refrigerant into tubing that circulates the fluid.

The biggest problem Lucas faced in his design was eliminating shock waves that formed in the cavity, wasting power as heat and thereby limiting compression. Lucas spent a year at Los Alamos National Laboratory working with acoustic and engine-cycle expert Gregory W. Swift to find a solution. "The trick was the geometric design of the resonator," Lucas points out. The shape made the higher harmonics that caused the shock waves to interfere with one another, leaving only the fundamental frequency in the cavity.

To an industry known for extreme caution in introducing new technology, a radically redesigned compressor may seem risky. Nor have manufacturers forgotten General Electric's debacle with its rotary compressor in the 1980s. After rushing it into production, GE discovered the compressor was defective; the company spent an estimated \$450 million to recall the products. But Lucas is unperturbed. "It appeared to be more of a management problem rather than an engineering one," he notes.

Refrigerator companies may have few other alternatives if they cannot meet tough energy efficiency standards cost-effectively or find an HFC 134a-compatible oil. Manufacturers should be able to meet 1993 efficiency guidelines by tweaking existing parts. But the oil compatibility issue remains, well, sticky. One industry expert thinks manufacturers could end up shortening the warranted lifetimes of the machinery if the gumming is not solved. All the more reason, perhaps, to chill out with a new compressor. —*Philip Yam*

Refrigerating by Sound



The sonic compressor is driven like an ordinary loudspeaker. The oscillation creates in the resonator cavity a high-amplitude standing wave, which compresses the refrigerant gas. A one-way valve vents the gas to a condenser, which liquefies and cools it. The liquid circulates around the space to be cooled, evaporating and thus absorbing heat. The gas is then pulled back into the compressor to repeat the cycle.

Breaking through the acoustic shock barrier

A sonic boom produced by a jet aircraft can shatter windows and flatten structures. Such a shock wave, whether generated in the open air or inside a closed, gas-filled tube, represents a significant concentration of acoustic energy.

Nonetheless, the formation of a shock front also marks a limit on the amount of energy that can be pumped into a sound wave. Energy added to a shock wave would dissipate without increasing the wave's peak pressure.

It is possible, however, to evade that limit by generating sound waves inside specially shaped cavities that prevent the formation of shock fronts, says Timothy S. Lucas of MacroSonix Corp. in Richmond, Va.

Lucas and his coworkers have developed cavity resonators within which standing sound waves of extremely high energy can be produced. The gas pressure inside such a resonator can reach hundreds of pounds per square inch, making the technology useful for compressing gases and other industrial processes.

For a long time, it was widely considered impossible to achieve such high-energy densities and acoustic pressures, Lucas says. "Our technology unlocks the power of sound."

The researchers described their work this week at an Acoustical Society of America meeting held in San Diego.

"The key is the shape of the resonator," says Gregory W. Swift of the Los Alamos (N.M.) National Laboratory. At high energies, shock waves form within cylindrical tubes but not inside tapered, streamlined cavities of just the right geometry.

Simply driving the new resonator back and forth at a frequency that depends on the size of the cavity and the type of gas it contains will produce a high-energy sound wave. As the oscillating pressure inside the cavity increases, gas molecules speed up, eventually traveling at about one-half the speed of sound.

The extreme pressure fluctuations in the resonator are analogous to surface waves on an imaginary lake, 1 kilometer deep, that shoot to a height of several kilometers and dip down to within a few hundred meters of the lake bottom, Swift says. Conversational sound, in contrast, corresponds to water waves about 1 millimeter high, and sound painful to the human ear is analogous to a wave height of 10 centimeters to 1 meter.

The new technology, known as resonant macrosonic synthesis, would offer a number of advantages if used in acoustic compressors for household refrigerators, in small turbines for generating electricity, and in chambers for separating, agglomerating, levitating, mixing, or pulverizing materials, Lucas says.

An acoustic compressor, for instance, eliminates the need for moving parts, such as pistons, connecting rods, crankshafts, and bearings, and thus for lubricating oil, he notes. An appliance manufacturer is already working with the technology to develop reliable, durable, energy-efficient refrigerators and air conditioners.

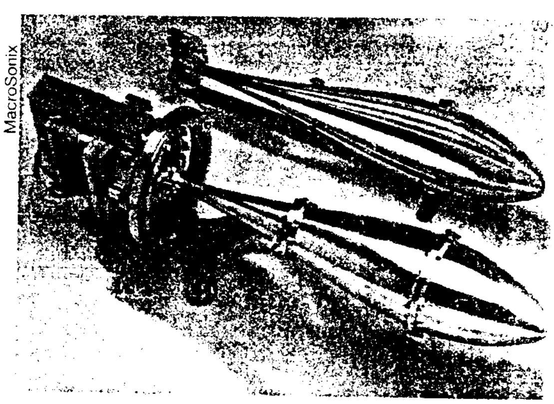
"You can use any refrigerant you want," Lucas says. "Because we have a simple, empty cavity, you don't have to worry about chemical incompatibility between the lubricating oil and the refrigerant."

Several scientific questions warrant further investigation. Lucas and his coworkers want to understand more fully the turbulence that accompanies the waves inside a resonator. "A large part of the dissipated energy goes into turbulence," Lucas says. "As we learn how to reduce turbulence, the energy efficiency of the machine would increase, perhaps double."

The researchers would also like to develop more accurate models of the acoustic effects that occur inside a resonator.

"This has turned out to be very interesting scientifically," Swift comments. "It's great stuff." —*I. Peterson*

MacroSonix



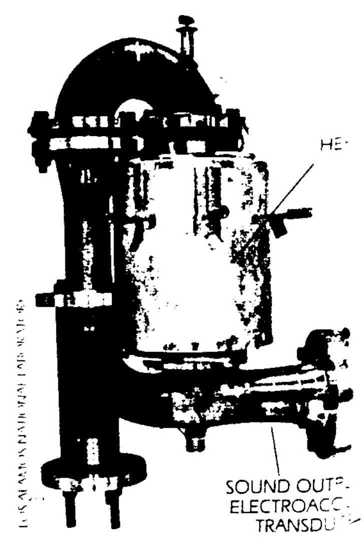
To generate a large-amplitude standing sound wave inside a specially shaped, gas-filled cavity, a motor (front, left) drives the elongated, tapered metal container, about 0.5 meter long, back and forth at a frequency of approximately 600 cycles per second. The cavity itself (shown in cutaway at back) contains no mechanical parts.

Engine Has No Moving Parts

An engine with no moving parts has been developed by scientists at the Los Alamos National Laboratory in New Mexico.

Called a thermoacoustic Stirling engine, it works by using externally applied heat to create traveling acoustic waves in helium. The resulting pressure fluctuations cause a resonator to vibrate at 80 Hz. This "sound" is then converted into electricity by a microphonelike electro-acoustic transducer.

The 700-watt prototype (right) is as efficient as an automobile engine. **PM**



 p. 30

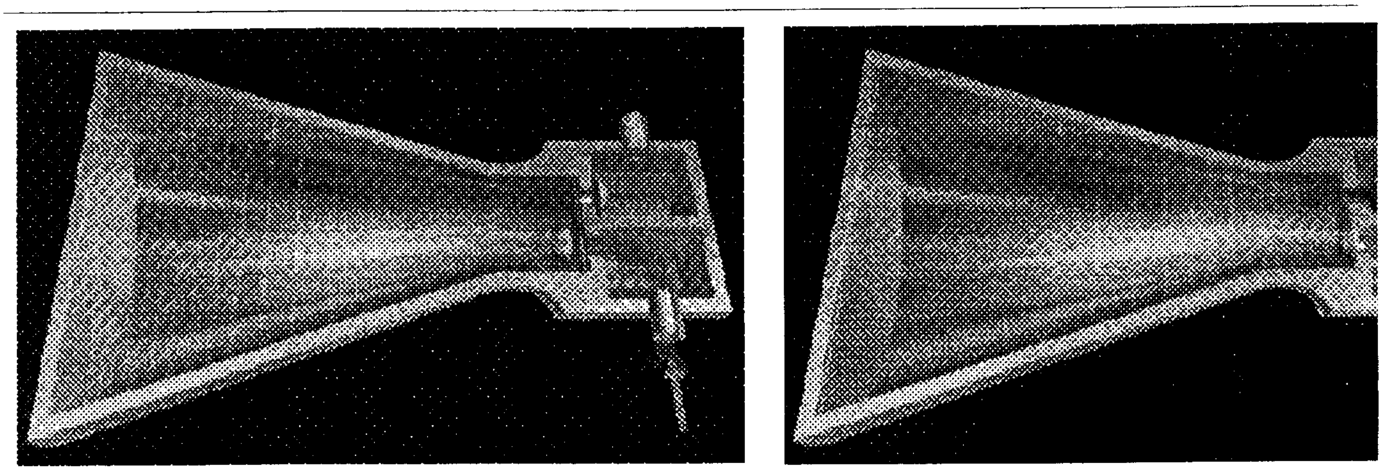
SCIENCE MAGAZINE

PHYSICS:

Cool Sounds at 200 Decibels

Dana Mackenzie

The loudest controlled sounds ever made by humans were produced earlier this month--not by a rock band, but by a physicist. At the Acoustical Society of America meeting in San Diego, Timothy Lucas of MacroSonix Corp. in Richmond, Virginia, demonstrated a new "acoustic compressor" that uses ultraintense sound waves to do the work of a mechanical pump. The technology may soon be used in everyday appliances such as refrigerators and air conditioners.



Sound concept. Cycles of low and high pressure driven by sound can draw a fluid into a compressor *(left)* and expel it at high pressure.

TIMOTHY LUCAS *ET AL.*/MACROSONIX CORP.

The idea of the compressor is simple: You shake a can back and forth to create vibrations in the air inside. Just as a child can produce huge waves in a bathtub by sloshing back and forth at just the right rate (a phenomenon called resonance), the air vibrations become especially intense if the can is agitated at a certain frequency. But the water in the child's bathtub will splash out if the waves start to crest. For acoustical engineers, the analogous problem is shock waves, which dissipate the sound energy as heat. By making his compressor just the right shape--essentially that of a bowling pin--Lucas was able to keep the shock waves from forming, even as the can vibrated at about 600 times a second.

How loud are the resulting sounds? The pain threshold is about 120 decibels, and a jet engine produces 150 decibels. If you stand next to a sound of 165 decibels, it will ignite your hair. The sound waves inside Lucas's compressor are about 3000 times more powerful, or about 200 decibels. But because the can's own vibrations are much smaller than the vibrations of the air, on the outside it

sounds just like an ordinary compressor.

The intense sound waves oscillate between low and high pressure in certain regions; with the help of valves that open and close at the right moments, these pressure differences can suck gas into the compressor and shoot it out at high pressure. Lucas's compressor could be especially useful for refrigerators and air conditioners, which work by compressing a refrigerant--traditionally a chlorofluorocarbon. Steve Garrett, a physicist at Pennsylvania State University in University Park, explains that some of the ozone-sparing refrigerants now being used break down in the oil that lubricates a conventional compressor. But Lucas's compressor has no moving parts inside and therefore requires no lubrication. MacroSonix has already signed a licensing agreement with an appliance manufacturer.

Other specialists in acoustics call Lucas's compressor a breakthrough. "What Timothy Lucas has done is shift the debate from whether acoustic compression can be done to who can do it better," says Garrett. Lucas himself thinks his sound waves will ultimately find many other roles. "Electromagnetic waves have been commercialized for over 100 years," he says, "but the commercial application of sound waves has only scratched the surface."

Dana Mackenzie is a science and mathematics writer in Santa Cruz, California.

Volume 278, Number 5346 Issue of 19 December 1997, p 2060

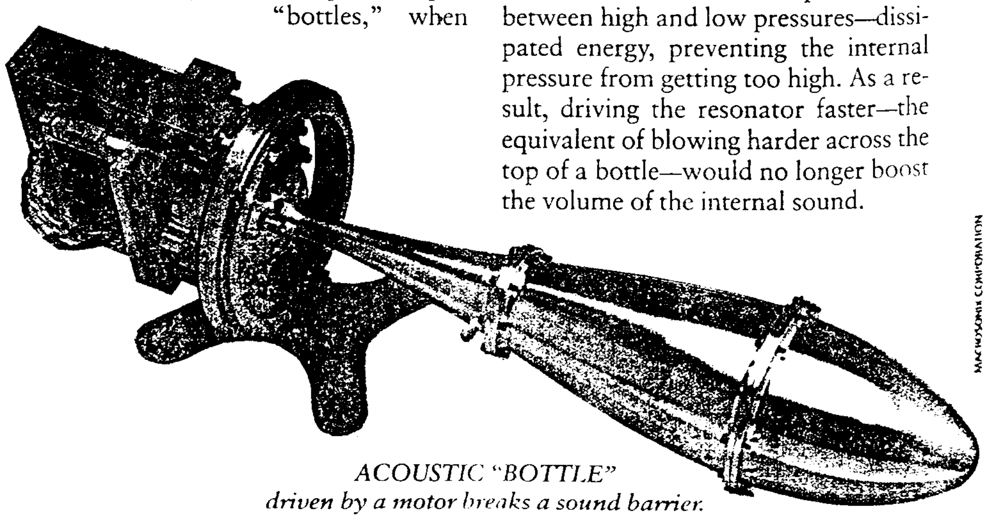
ACOUSTICS

BOOM BOX

A resonator boosts sound pressures to new highs

Blowing across the lip of a bottle to produce that satisfying hum would not seem to be the basis for new discoveries. But that is essentially what Timothy S. Lucas claims he has made. Reporting at the Acoustical Society of America meeting last December, the founder and president of MacroSonix Corporation in Richmond, Va., says his torpedo-shaped "bottles," when shaken back and forth hundreds of times a second, can create standing sound waves within them that pack energy densities 1,000 times greater than those previously achieved in acoustics. The process, which Lucas calls "resonant macrosonic synthesis," can produce pressures exceeding 3.5 million pascals (500 pounds per square inch), more than enough for industrial applications such as compressing and pumping.

The key is the shape of the bottle, or resonator. In the past, resonators were often cylindrical, and shock waves formed inside them if they vibrated too quickly. A shock wave—a compression wave that delineates a sharp boundary between high and low pressures—dissipated energy, preventing the internal pressure from getting too high. As a result, driving the resonator faster—the equivalent of blowing harder across the top of a bottle—would no longer boost the volume of the internal sound.



ACOUSTIC "BOTTLE"
driven by a motor breaks a sound barrier.

While at Los Alamos National Laboratory in 1990, Lucas studied how shock waves could be broken down into higher-frequency components, or harmonics. He realized that for resonant waves, the shape of the cavity was the critical factor. Lucas's resonators, which can also be in the shape of bulbs and cones, cause the harmonics to add up slightly out of step with one another. As a result, there are no overly sudden changes in pressure that lead to shock fronts. Without shock formation, the intensity of sound waves could build up, reaching amplitudes not previously possible.

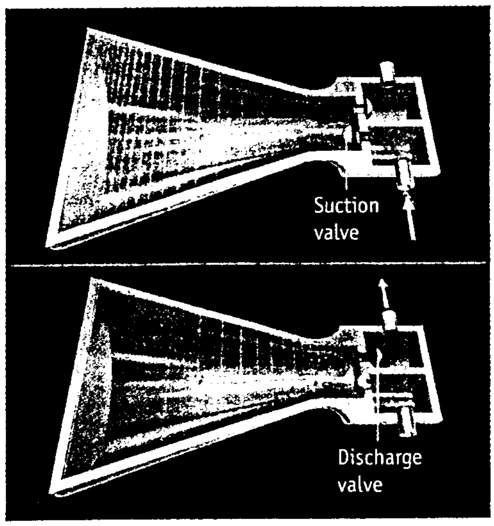
Currently Lucas and his colleagues are modeling the acoustics within the cavity: some of the turbulence inside robs energy from the sound wave. Still, the resonator has generated enough sonic power to interest a major appliance manufacturer, which has a license to incorporate the resonator as a compressor in household refrigerators. —*Philip Yam*

p. 28

Sound Thinking

A PATENTED TECHNOLOGY called Resonant Macrosonic Synthesis, or RMS, harnesses sound waves to do high-powered mechanical work—something experts in the field of acoustics once thought impossible. Previously, researchers who had tried to create high-energy sound waves had instead formed shock waves, which dissipate energy.

The problem, according to physicist Tim Lucas, is that earlier researchers tried to create sound waves inside closed cylindrical cavities, called resonators. Lucas instead uses conical and bomb-shaped resonators. The geometry of these resonators defeats shock-wave formation, making it possible to create standing sound waves with energy densities 1,600 times greater than ever before achieved acoustically.



A standing sound wave inside this compressor causes pressure to oscillate between high and low. When pressure at the small end is low (blue), the suction valve opens and low-pressure gas (green) enters. When pressure is high (red), the discharge valve opens and high-pressure gas is released.

Lucas, who worked on acoustic problems at Los Alamos National Laboratory and now heads the MacroSonix Corp. in Richmond, Virginia, says the RMS technology generates sound waves with enough pressure to replace mechanical compressors in a wide variety of applications. For example, an RMS compressor could be used in household refrigerators and air conditioners. Not only would an acoustic compressor be lubricant-free, with no moving parts to maintain, it could also be 30 to 40 percent more efficient than a conventional compressor, Lucas claims. Purdue University engineering professor Luc Mongeau, who is an expert in acoustical refrigeration, calls RMS an "amazing breakthrough."

But RMS sound technology isn't just for compressors, says Lucas. For example, the technology could also be used to separate, mix, or pulverize materials in the chemical and pharmaceutical industries.—*A.F.*

1998 Discover Awards

SOUND POWER

MacroSonix's Resonant Sound Technology

INNOVATOR: TIM LUCAS

Imagine a compressor in your refrigerator with no pistons, crankshafts, or lubricated bearings. Instead, all the work is done by sound waves bouncing around in an empty cavity.

When this idea first began bouncing around Tim Lucas's head ten years ago, his fellow physicists told him it would never work. Sound waves, they pointed out, can store only a relatively small amount of energy before turning into jagged shock waves that dissipate any added energy as heat. At least that's what happens when a wave travels through the open air, or through a cylindrical "wave guide." Undaunted, Lucas experimented and found that by shaping the sound chamber, or resonator, into something like a cone or a bulb, he could keep shock waves from forming. "Most of the research had been done in a simple cylindrical tube, and it turns out that's the one resonator guaranteed to give you a shock wave," says Lucas. "There's an infinite family of resonators that can give you non-shocked waves." In his technology, which he calls Resonant Macrosonic Synthesis, sound waves store thousands of times more energy than previously thought possible.

Lucas, who started his own company, MacroSonix Corp. in Richmond, Virginia, to develop RMS, has licensed it to one company (he won't say which) for refrigerator compressors—the part that compresses and circulates the coolant. The coolant passing through the cavity would be compressed when it encounters the high-pressure portion of the wave. Other applications might include cooling computer chips; "micronizaton," which is the pulverizing of particles down to microscopic size; and filtering out particles from factory exhaust (the sound waves would cause the particles to clump together). "We believe RMS is a new primary technology, something that functions at a fundamental level of physics," says Lucas.

A SOUND RESONATING CAVITY, AND A CUTAWAY VERSION (REAR).